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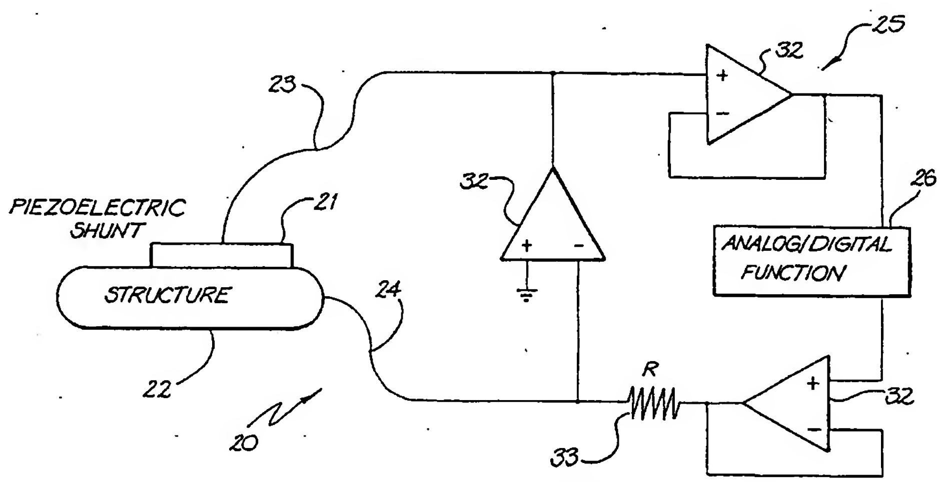
(54) Title: AN IMPEDANCE SYNTHESISING ARRANGEMENT, AN IMPROVED VIBRATIONAL DAMPING APPARATUS AND A METHOD FOR DERIVING A DIGITAL SIGNAL PROCESSING ALGORITHM



(57) Abstract: A vibrational damping apparatus (20) with piezoelectric component (21) is engaged with a structure (22) which is to be vibrationally damped. The piezoelectric component (21) has a pair of electrical terminals (23) and (24) for communication of a voltage across the piezoelectric material. The impedance synthesizing arrangement (25) may include an analog or digital means (26) for implementing a transfer function. The predefined relationship provided by the transfer function is adapted to synthesize a circuit for passive shunting of the piezoelectric material (21). The terminals (23, 24) of the piezoelectric component (21) are electrically connected to the impedance synthesizing arrangement (25) such that input voltage (11) is provide by the piezoelectric material (21) in response to vibration of the structure (22) and the output current (12) is fed to piezoelectric material (21) so as to vibrationally damp the structure.

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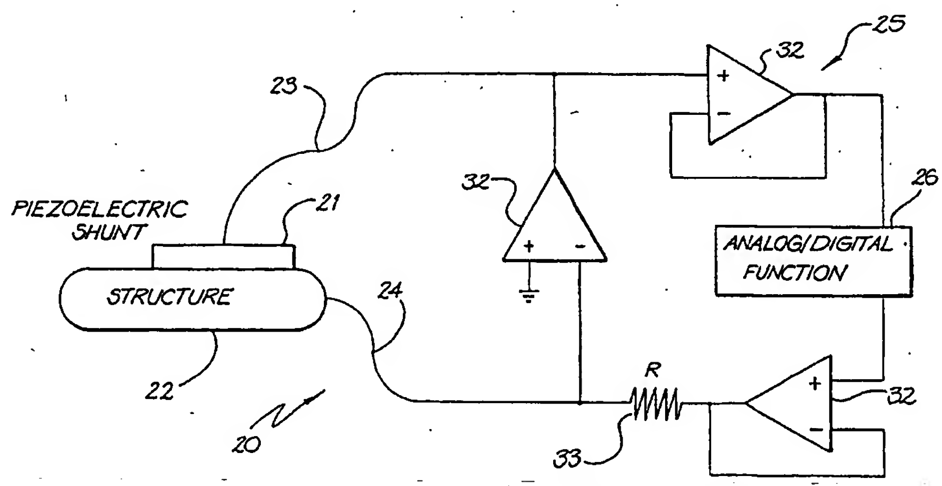

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Title:

An Impedance Synthesising Arrangement, An Improved Vibrational Damping Apparatus and a Method for Deriving a Digital Signal Processing Algorithm.

Technical Field of the Invention:

The invention has been developed primarily for use in the field of piezoelectric vibrational damping and will be described hereinafter with reference to this application. However, it will be appreciated that the invention is not limited to this particular field of use.

Background of the Invention:

Prior art analog circuitry utilising components such as resistors, capacitors, inductors etc can suffer from a number of disadvantages such as excessive circuit complexity, component cost and suboptimal performance if the temperature of the components exceeds ordinary operating limits.

It has been appreciated by the inventors of the present invention that such disadvantages apply to prior art vibrational damping arrangements utilising passively shunted piezoelectric material. An example of such prior art is provided by the journal article entitled "Damping of Structural Vibrations with Piezoelectric Materials and Passive Electrical Networks" (N. W. Hagood and A. von Flotow, *Journal of Sound and Vibration*, 146 (2) 243-268), the contents of which are hereby incorporated in their entirety by reference.

It will be appreciated by those skilled in the art that a further disadvantage associated with the damping arrangement disclosed by the Hagood article is

that only one of the modes of vibration of the structure is damped. This is partially addressed by US Patent No. 5,783,898 which discloses an arrangement which can damp more than one mode of vibration. However it has been appreciated by the inventors that there are a number of problems with this technique, the foremost being the complexity and size of the circuit required to implement the total impedance. Typically the shunt circuits contain up to 48 opamps for a three mode circuit and require large inductor values, which may be up to hundreds of Henries. The more modes that are shunted, the greater the number of opamps required. Virtual grounded inductors and virtual floating inductors (Riordan gyrators) are required to implement the grounded and floating inductor elements. These circuits are typically poor representations of ideal inductors, are large in size, and are sensitive to component tolerances and non-ideal characteristics.

Piezoelectric patches are capable of generating hundreds of volts for moderate structural excitations and this requires the entire shunt circuit to be constructed from high voltage components, with significant associated component cost. Further voltage limitations arise due to the internal gains in the virtual inductor circuits.

Further, assuming ideal components, there is a fundamental limitation in using the blocking circuit technique taught by said US Patent. It is desired to design each parallel shunt branch so that it is only significant over the narrow frequency band of its corresponding structural resonant mode. For two mode damping it is sufficient to assume the current blocker circuit has little effect on the dynamics of the desired R-L shunt circuit around the modal frequency and that the branch will not effect other branches designed for higher resonant

frequencies. For this case a 3 dB to 6 dB damping performance penalty is encountered. For a three mode circuit, some or all of the branches would require a series combination of three current blocker circuits This would severely limit the damping performance at each mode in comparison with single mode damping.

Finally, the shunt circuit disclosed in US Patent No. 5,783,898 must be accurately tuned to each of the frequencies of the modes of vibration that are to be damped. As the prior art circuit componentry heats up, the circuit may effectively be de-tuned due to suboptimal performance of the components, and the inherent tolerances in specification of components such as capacitors and resistors. This may result in significantly less effective damping of one or more modes of structural vibrations.

Any discussion of the prior art throughout the specification should in no way be considered as an admission that such prior art is widely known or forms part of common general knowledge in the field.

Summary of the Invention:

It is an object of the present invention to overcome or ameliorate at least one of the disadvantages of the prior art, or to provide a useful alternative.

According to a first aspect of the invention there is provided an impedance synthesising arrangement adapted to accept an input voltage and to provide a corresponding output current according to a predefined relationship between the input voltage and the output current.

According to a second aspect of the invention there is provided an impedance synthesising arrangement adapted to accept an input current and to provide a corresponding output voltage according to a predefined relationship between the input current and the output voltage.

Preferably in either of the preceding aspects of the invention, the predefined relationship is adapted to synthesise a network of electrical components.

In one preferred embodiment the impedance synthesising arrangement includes a digital signal processor and in another embodiment it includes analog circuitry to define the said voltage - current relationship.

According to a third aspect of the invention there is provided a vibrational damping apparatus including:

a piezoelectric component for engagement with a structure to be vibrationally damped, the piezoelectric component having a pair of electrical terminals for communication of a voltage across the piezoelectric material; and

an impedance synthesising arrangement in accordance with the first aspect of the invention wherein said predefined relationship is adapted to synthesise a shunting circuit;

the terminals of said piezoelectric component being electrically connected to the impedance synthesising arrangement such that the input voltage is provided by said piezoelectric material in response to vibration of the structure and the output current is fed to said piezoelectric material so as to vibrationally damp said structure.

According to a fourth aspect of the invention there is provided a vibrational damping apparatus including:

a piezoelectric component for engagement with a structure to be vibrationally damped, the piezoelectric component having a pair of electrical terminals for communication of a voltage across the piezoelectric material; and

an impedance synthesising arrangement in accordance with the second aspect of the invention wherein said predefined relationship is adapted to synthesise a shunting circuit;

the terminals of said piezoelectric component being electrically connected to the impedance synthesising arrangement such that the input current is provided by said piezoelectric material in response to vibration of the structure and the output voltage is fed to said piezoelectric material so as to vibrationally damp said structure.

According to another aspect of the invention there is provided a method for deriving a digital signal processing algorithm to be utilised in the impedance synthesising arrangement described above, said method including the following steps:

1) designing a shunt circuit having a plurality of impedances which, in combination with any inherent capacitance of the piezoelectric component, would give desired vibrational damping properties;

2) converting said shunt circuit into a transfer function block diagram; and

3) forming a digital signal processing algorithm from said block diagram.

Preferably the step of "forming a digital signal processing algorithm from said block diagram" is performed by a graphical compilation package, such as

the real-time workshop for MATLAB, to be implemented on the rapid proto-type system dSpace.

Brief Description of the Drawings:

Preferred embodiments of the invention will be described, by way of example only, with reference to the accompanying drawings, in which:

figure 1 is a circuit diagram illustrating the functionality of a first embodiment of an impedance synthesising arrangement;

figure 2 is a circuit diagram illustrating a vibrational damping apparatus utilising the first embodiment of the impedance synthesising arrangement;

figure 3 is a circuit diagram illustrating a piezoelectric material connected to a first shunt circuit;

figure 4 is a circuit diagram illustrating a piezoelectric material connected to a second shunt circuit;

figure 5 is a circuit diagram illustrating a piezoelectric material connected to a third shunt circuit;

figure 6 illustrates a parallel network of impedances and the equivalent transfer function block diagram;

figure 7 illustrates a series network of impedances and the equivalent transfer function block diagram;

figure 8 is a graph illustrating the results of an experiment to compare undamped vibrations with vibrations damped using a preferred embodiment of the improved vibrational damping apparatus;

figure 9 is a circuit diagram illustrating the functionality of a second embodiment of an impedance synthesising arrangement; and

figure 10 is a circuit diagram illustrating a shunt circuit being synthesised by the second embodiment of an impedance synthesising arrangement for connection to a piezoelectric material.

Detailed Description of Preferred Embodiments of the Invention:

Referring to the drawings, the functionality of a first embodiment of an impedance synthesising arrangement is illustrated in figure 1. The impedance synthesising arrangement, shown schematically as reference numeral 10, is adapted to accept an input voltage 11 and to provide a corresponding output current 12 according to a predefined relationship between the input voltage and the output current. In mathematical terms:

$$i_z(t) = f\{v(t)\}$$

where the function f is the "predefined relationship" which can be made to synthesise a network of physical components, for example by fixing i_z to be the output of a linear transfer function of v_z, ie,

$$I_z(s) = Y(s)\,V_z(s),$$

where $Y(s) = 1 / Z(s)$

and $Z(s)$ is the impedance to be seen from the terminals.

Figure 9 illustrates another embodiment of an impedance synthesising arrangement 40 adapted to accept an input current i_z 41 and to provide a corresponding output voltage v_z 42 according to a predefined relationship between the input current i_z 41 and the output voltage v_z 42. In mathematical terms:

$$v_z(t) = f\{\, i_z(t) \,\}$$

where the function f is a "predefined relationship" which can be made to synthesise a network of physical components as discussed above.

Referring now to the vibrational damping apparatus 20 shown in figure 2, the piezoelectric component 21 is engaged with a structure 22 which is to be vibrationally damped. In some embodiments the piezoelectric component 21 is bonded to the structure 22 with a thin layer of adhesive epoxy, although other methods of engagement may be used. Clearly the piezoelectric material should be placed at or near a vibrational antinode for maximum vibrational damping efficiency, although other positions may suffice for certain purposes. The piezoelectric component 21 has a pair of electrical terminals 23 and 24 for communication of a voltage across the piezoelectric material.

The impedance synthesising arrangement 25 utilised in the vibrational damping arrangement 20 may include an analog or preferably digital means 26 for implementing a transfer function. The predefined relationship provided by the transfer function is adapted to synthesise a circuit for passive shunting of the piezoelectric material 21.

The terminals 23, 24 of the piezoelectric component 21 are electrically connected to the impedance synthesising arrangement 25 such that the input voltage 11 is provided by the piezoelectric material 21 in response to vibration of the structure 22 and the output current 12 is fed to the piezoelectric material 21 so as to vibrationally damp the structure.

The means 26 for implementing a transfer function which is part of the impedance synthesising arrangement 25 is preferably a digital signal processor, although in some embodiments may be an analog circuit.

Some of the possible shunting circuits which may be synthesised by the impedance synthesising arrangement 25 are shown in figures 3, 4 and 5. More particularly, the circuit which is synthesised by the impedance synthesising arrangement 25 is the circuit minus the piezoelectric component 21 (also labelled PZT), to which the impedance synthesising arrangement 25 is electrically connected. These shunting circuits, in combination with any inherent capacitance 30 of the piezoelectric component, are tailored to result in an impedance experienced by the piezoelectric material 21 which has a first local maximum at a frequency substantially equal to a first resonant frequency of a first mode of vibration of the structure. In other words, if one of the modes of vibration to be damped occurs at, say, 74.6 Hz then the impedance experienced by the piezoelectric material 21 is tailored to have a local maximum at 74.6 Hz. If it is desired to vibrationally damp more than one mode, the shunting circuit synthesised by said impedance synthesising arrangement 25, in combination with any inherent capacitance 30 of the piezoelectric component 21, is tailored to result in an impedance which has local maxima at frequencies substantially equal to resonant frequencies of modes of vibration of the structure. For example, if two of the modes of vibration to be damped occur at, say, 74.6 Hz and 171.4 Hz, then the impedance experienced by the piezoelectric material 21 is tailored to have local maxima at 74.6 Hz and 171.4 Hz.

As can be best seen from figures 3, 4 and 5, the shunting circuits synthesised by the impedance synthesising arrangement 25, in combination with any inherent capacitance 30 of the piezoelectric component 21, have the effect of one or more L-C-R circuits 31, each tuned to resonate at different resonant frequencies of modes of vibration of the structure 22. More particularly, the

shunt circuits include a plurality of such L-C-R circuits 31, at least some of which are subject to blocking means 34 adapted to ensure that substantially only the respective resonant frequency is fed to each L-C-R circuit. In other words, if one L-C-R circuit 31 is tuned for a maximum impedance at, say, 74.6 Hz, then its blocking means 34 ensures substantially only that frequency is fed to that L-C-R circuit 31.

The blocking means 34 used in the circuits shown in figures 3, 4 and 5 are L-C circuits tuned to anti-resonate as required for blockage purposes.

As shown in figure 2, three opamps 32 and a resistor 33 electrically connect the impedance synthesising arrangement 25 to the piezoelectric component 21.

Figure 10 illustrates an alternative vibrational damping apparatus 43 which utilises the second embodiment of the impedance synthesising arrangement 40. V_c is a voltage controlled voltage source (non inverting amplifier) with gain G_2. The gains G_1 and G_2 are included to allow magnitude conditioning of the signals a(t) and b(t). This is useful to achieve a maximum signal to noise ratio if a digital signal processor is used to synthesize Z(s). Note that the signal a(t) is equivalent to the terminal current in $G_1 \frac{volts}{amp}$.

The terminal current $i_z(t)$ 44 is measured, in this case by sensing the voltage across resistor Rs 45, however in alternative embodiments other means such as a Hall Effect Sensor may be employed to measure the current 44.

$$I_z(s) = \frac{X(s) - Y(s)}{R_s}$$

It follows that:

$$I_s(s)=\frac{V_s(s)}{Z(s)}-\frac{I_s(s)R(s)}{Z(s)}$$

Thus we arrive at the relationship between terminal voltage and current:

$$Z_T(s)=\frac{V_z(s)}{I_s(s)}=Z(s)+R_s$$

The terminal impedance $Z_T(s)$ is equivalent to Z(s) + R_s where Z(s) is the impedance transfer function of some electrical network. In use, the terminals 46 and 47 are connectable to piezoelectric material in order to provide vibrational damping.

When using a digital signal processor 26 in the impedance synthesising means 25, it is preferable to develop an algorithm which can be used to ensure that the predefined relationship between the voltage 11 and the current 12 is as required. The preferred method for deriving a digital signal processing algorithm includes the following steps:

1) designing a shunt circuit (for example as illustrated in figures 3, 4 or 5) having a plurality of impedances which, in combination with any inherent capacitance 30 of the piezoelectric component 21, would give desired vibrational damping properties;

2) converting said shunt circuit into a transfer function block diagram (such as those illustrated in figures 6 and 7);

3) forming a digital signal processing algorithm from said block diagram.

It will be appreciated by those skilled in the art that combinations of the parallel and series arrangements shown in figures 6 and 7 may be used for circuits having combinations of parallel and series impedances.

The step of "forming a digital signal processing algorithm from said block diagram" is preferably performed by a graphical compilation package, such as the Real Time Workshop for MATLAB, although other methods are known to those skilled in the art.

An alternative to using the "block" method described in the preceding paragraphs is to compute from first principles an overall impedance for the shunt circuit, based upon an analysis of each of the separate components and their interrelationships. This approach has the disadvantage of increased computational difficulty, however has the advantage of not requiring a graphical compilation package or conversion of the shunt circuit into a transfer function block diagram.

Preferred embodiments of the vibrational damping apparatus were built for the purposes of testing. Two structural modes of vibration of a simply supported beam 22 were damped using a simulation of the shunt circuit shown in figure 3. The circuit shown in figure 2 was used to connect the synthetic impedance arrangement 25 to the piezoelectric component 21. The component values to damp two modes at 74.6 Hz and 171.4 Hz were:

$R_1 = 262.75\ k\Omega$;

$R_2 = 550.73\ k\Omega$;

$L_1 = 42\ H$;

$L_2 = 20.3\ H$;

$L_3 = 45.2$ H; and

$C_3 = 100$ nF.

The results are shown in figure 8 where the frequency response is measured from the applied structural excitation magnitude to structural velocity magnitude at a point. There was found to be a 20 dB and 18 dB reduction of the two resonant peaks. It was found experimentally that these results were considerably better than those obtained by using the technique taught by US Patent No. 5,783,898 with the same shunt circuit, even after extensive tuning to calibrate the virtual inductors and branch frequencies.

Additionally, the preferred embodiment of the vibrational damping apparatus 20 may be easily used to damp higher order modes by designing the appropriate damping network and implementing the algorithm derived therefrom.

Although the invention has been described with reference to specific examples, it will be appreciated by those skilled in the art that the invention may be embodied in many other forms.

CLAIMS:-

1. An impedance synthesising arrangement adapted to accept an input voltage and to provide a corresponding output current according to a predefined relationship between the input voltage and the output current.

2. An impedance synthesising arrangement adapted to accept an input current and to provide a corresponding output voltage according to a predefined relationship between the input current and the output voltage.

3. An impedance synthesising arrangement according to claim 1 or 2 wherein said predefined relationship is adapted to synthesise a network of electrical components.

4. A vibrational damping apparatus including:

a piezoelectric component for engagement with a structure to be vibrationally damped, the piezoelectric component having a pair of electrical terminals for communication of a voltage across the piezoelectric material; and

an impedance synthesising arrangement according to claim 1 wherein said predefined relationship is adapted to synthesise a shunting circuit;

the terminals of said piezoelectric component being electrically connected to the impedance synthesising arrangement such that the input voltage is provided by said piezoelectric material in response to vibration of the structure and the output current is fed to said piezoelectric material so as to vibrationally damp said structure.

5. A vibrational damping apparatus including:

a piezoelectric component for engagement with a structure to be vibrationally damped, the piezoelectric component having a pair of electrical terminals for communication of a voltage across the piezoelectric material; and

an impedance synthesising arrangement according to claim 2 wherein said predefined relationship is adapted to synthesise a shunting circuit;

the terminals of said piezoelectric component being electrically connected to the impedance synthesising arrangement such that the input current is provided by said piezoelectric material in response to vibration of the structure and the output voltage is fed to said piezoelectric material so as to vibrationally damp said structure.

6. A vibrational damping apparatus according to claim 4 or 5 wherein said impedance synthesising arrangement includes a digital signal processor.

7. A vibrational damping apparatus according to claim 4 or 5 wherein said impedance synthesising arrangement includes an analog low pass filter with gain.

8. A vibrational damping apparatus according to any one of claims 3 to 7 wherein said shunting circuit synthesised by said impedance synthesising arrangement, in combination with any inherent capacitance of the piezoelectric component, results in an impedance which has a first local

maximum at a frequency substantially equal to a first resonant frequency of a first mode of vibration of the structure.

9. A vibrational damping apparatus according to claim 8 wherein said shunting circuit synthesised by said impedance synthesising arrangement, in combination with any inherent capacitance of the piezoelectric component, results in an impedance which has local maxima at frequencies substantially equal to resonant frequencies of modes of vibration of the structure.

10. A vibrational damping apparatus according to any one of claims 4 to 9 wherein said shunting circuit synthesised by said impedance synthesising arrangement, in combination with any inherent capacitance of the piezoelectric component, has the effect of one or more L-C-R circuits, each tuned to resonate at different resonant frequencies of modes of vibration of the structure.

11. A vibrational damping apparatus according to any one of claims 4 to 10 wherein three opamps and a resistor electrically connect the impedance synthesising arrangement of claim 1 to said piezoelectric component in accordance with the circuit shown in figure 2.

12. A vibrational damping apparatus according to any one of claims 4 to 10 wherein the impedance synthesising arrangement of claim 2 is connected to said piezoelectric component in accordance with the circuit shown in figure 10.

13. A vibrational damping apparatus according to any one of the preceding claims wherein said shunt circuit includes a plurality of L-C-R circuits, each being tuned to resonate at different resonant frequencies of modes of vibration of the structure, at least some of said L-C-R circuits being subject to blocking means adapted to ensure that substantially only the respective resonant frequency is fed to each L-C-R circuit.

14. A method for deriving a digital signal processing algorithm to be utilised in the impedance synthesising arrangement of any one of claims 1 to 13, said method including the following steps:

1) designing a shunt circuit having a plurality of impedances which, in combination with any inherent capacitance of the piezoelectric component, would give desired vibrational damping properties;

2) converting said shunt circuit into a transfer function block diagram;

3) forming a digital signal processing algorithm from said block diagram.

15. A method for deriving a digital signal processing algorithm according to claim 14, wherein said step of "forming a digital signal processing algorithm from said block diagram" is performed by a graphical compilation package.

16. An impedance synthesising arrangement substantially as herein described with reference to any one of the disclosed embodiments and their associated drawings.

17. A vibrational damping apparatus as herein described with reference to any one of the disclosed embodiments and their associated drawings.

18. A method for deriving a digital signal processing algorithm as herein described with reference to any one of the disclosed embodiments and their associated drawings.

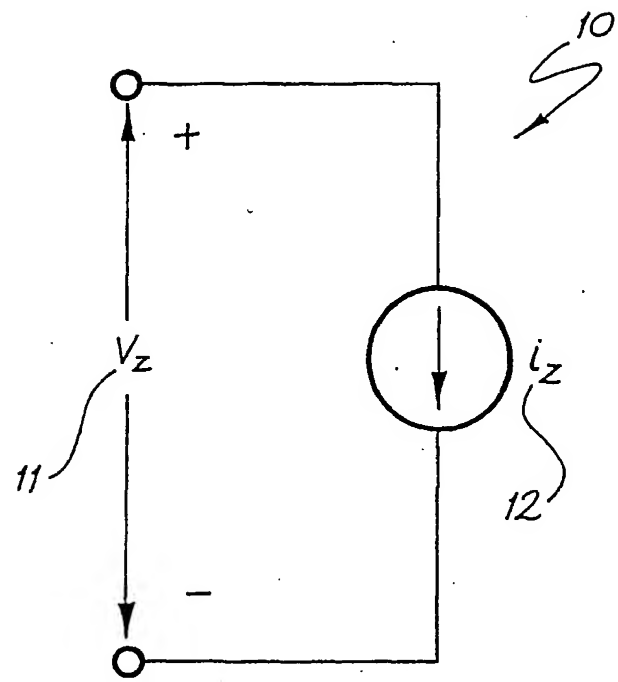
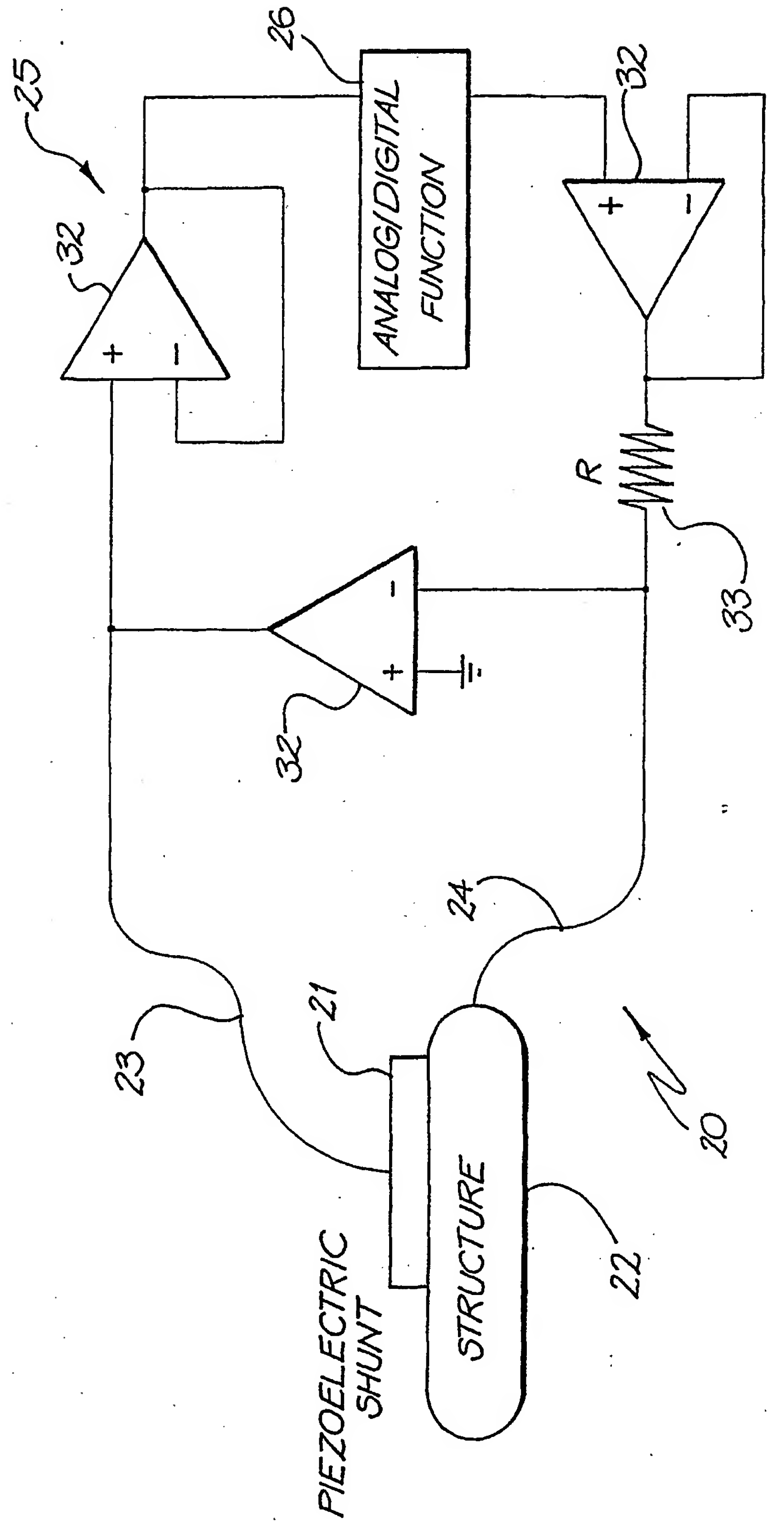


FIG. 1


25
26
ANALOG/DIGITAL
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PIEZOELECTRIC
SHUNT
STRUCTURE
22

FIG.2

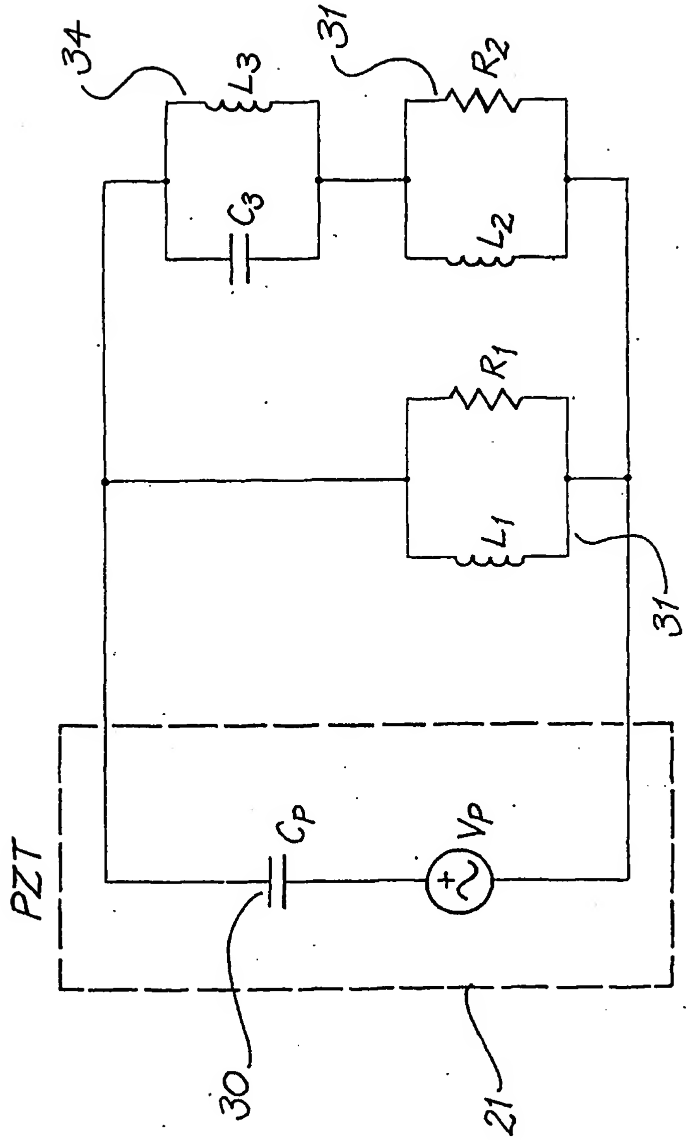


FIG. 3

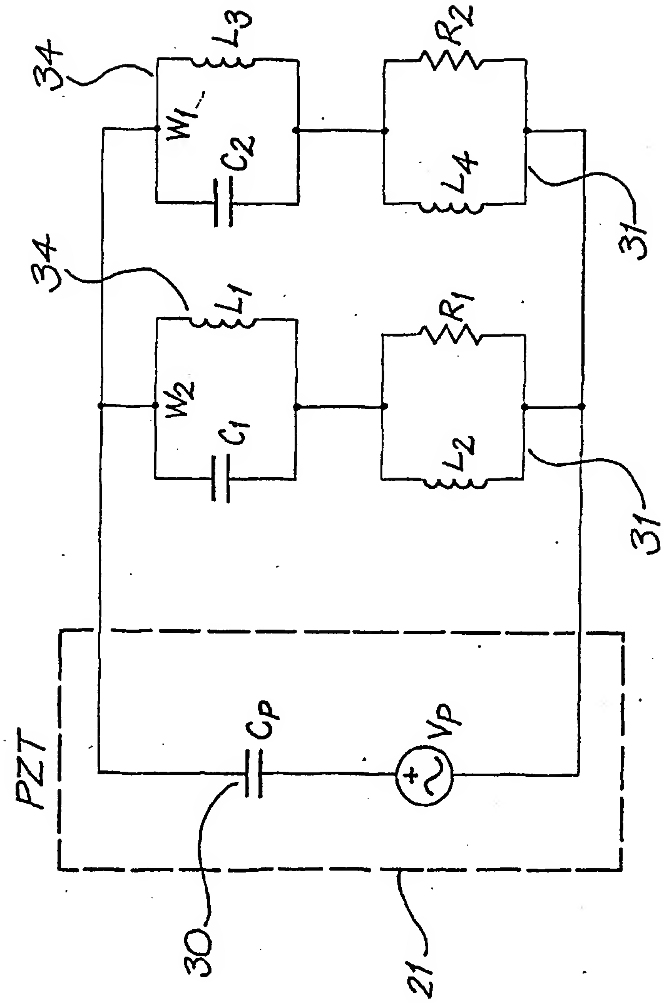


FIG.4

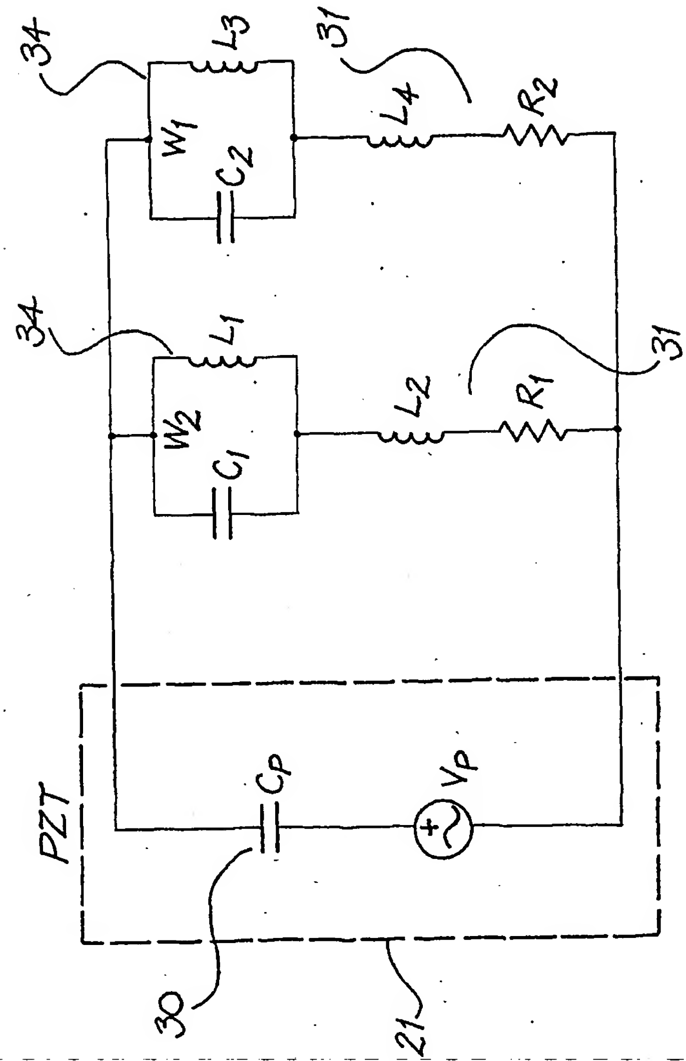


FIG. 5

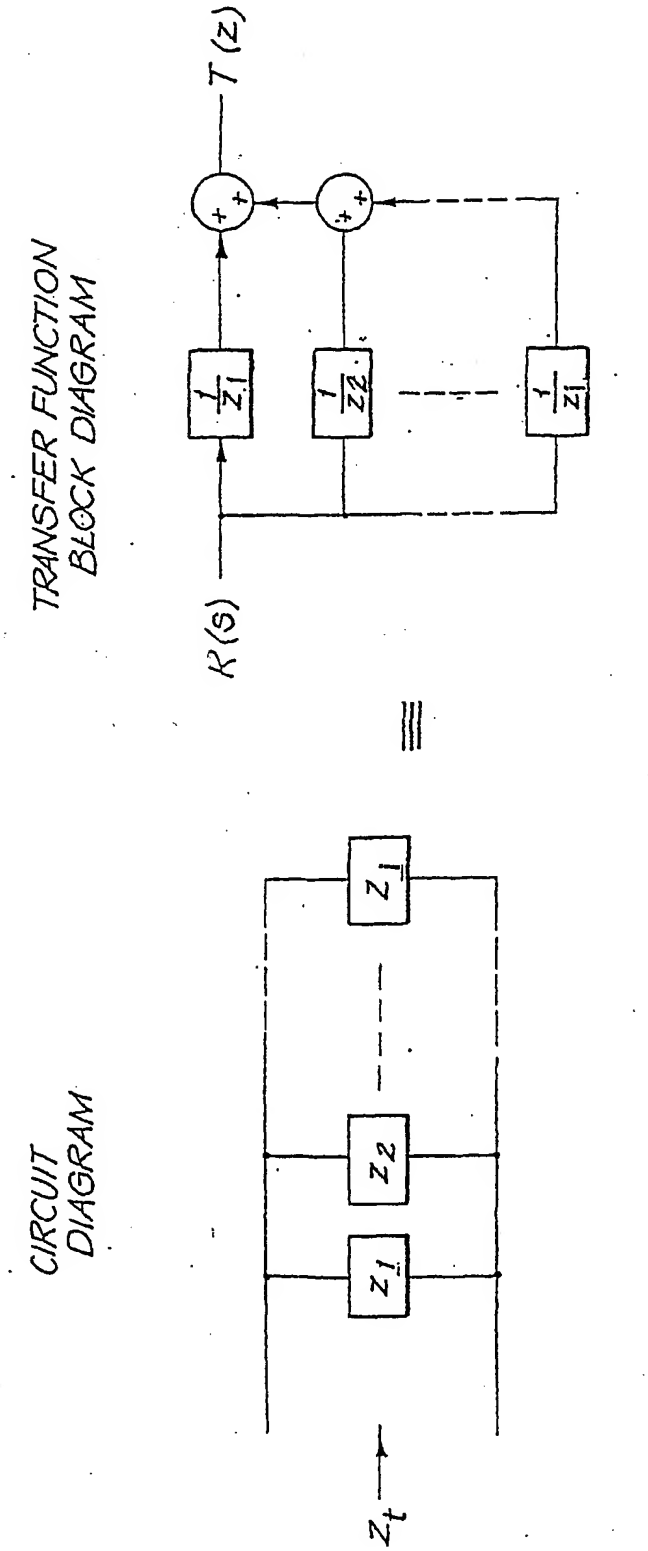
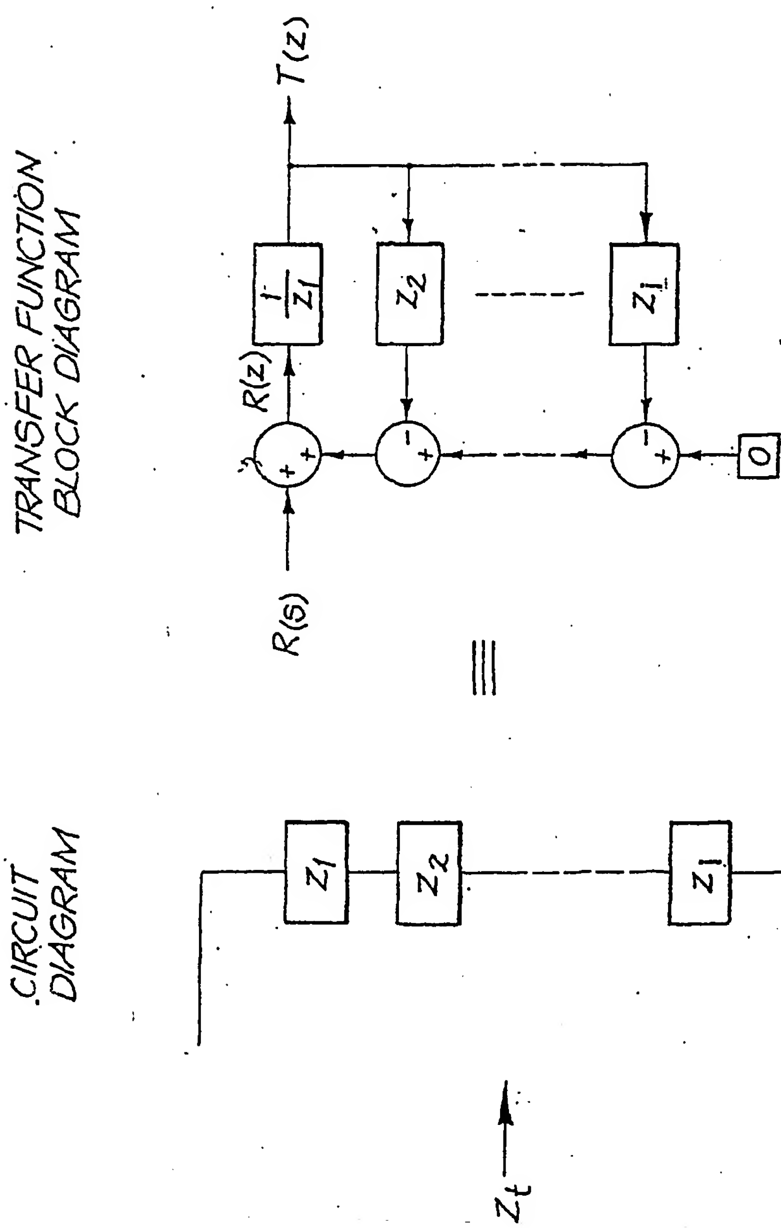

TRANSFER FUNCTION BLOCK DIAGRAM
R(s)
T(z)
1/Z1
1/Z2
1/Zi
CIRCUIT DIAGRAM
Zt
Z1
Z2
Zi

FIG.6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Zl
0
CIRCUIT
DIAGRAM
Z1
Z2
Zl
Zt

FIG.7

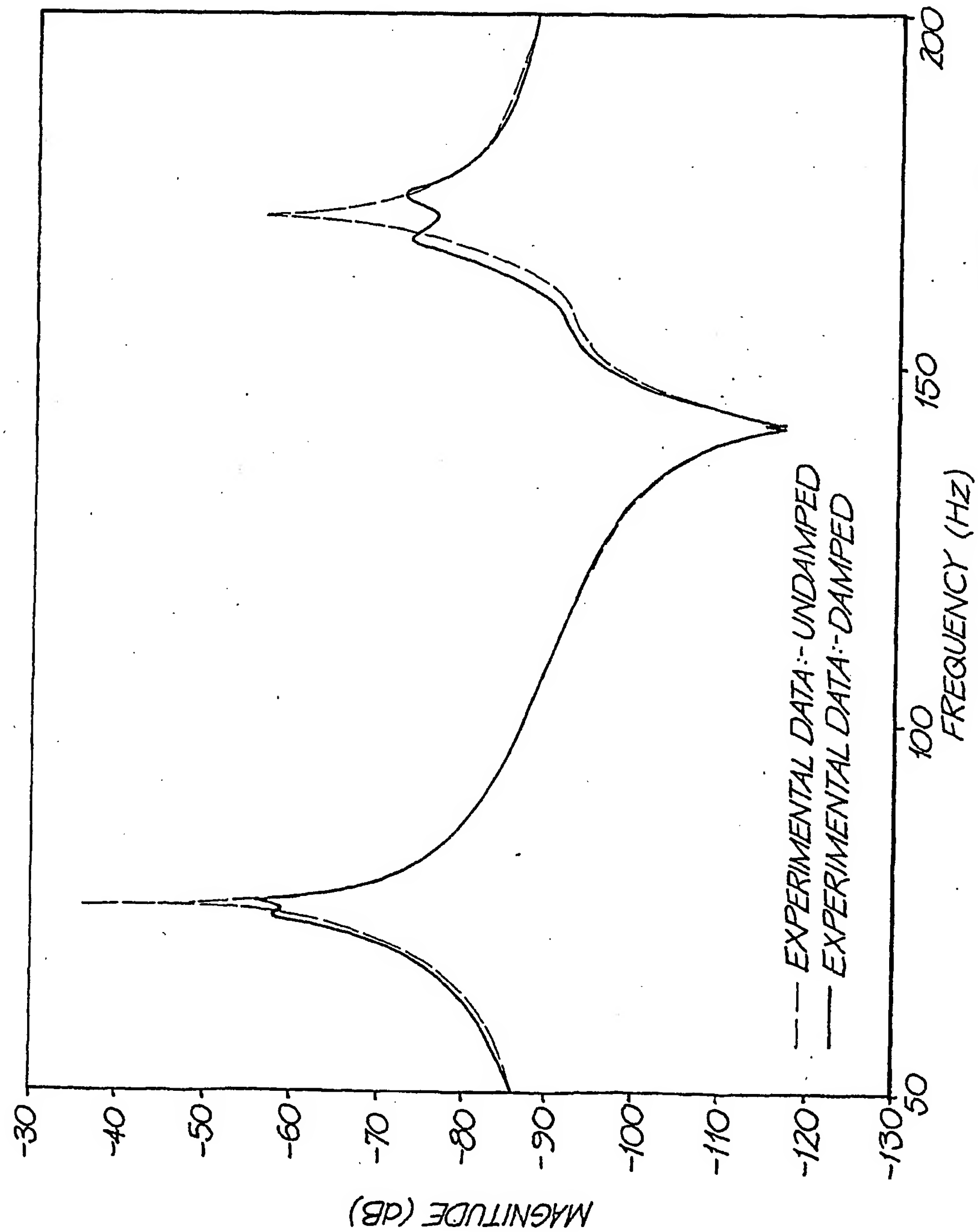
-30
-40
-50
-60
-70
-80
-90
-100
-110
-120
-130
MAGNITUDE (dB)
50
100
150
200
FREQUENCY (Hz)
EXPERIMENTAL DATA:-UNDAMPED
EXPERIMENTAL DATA:-DAMPED

FIG. 8

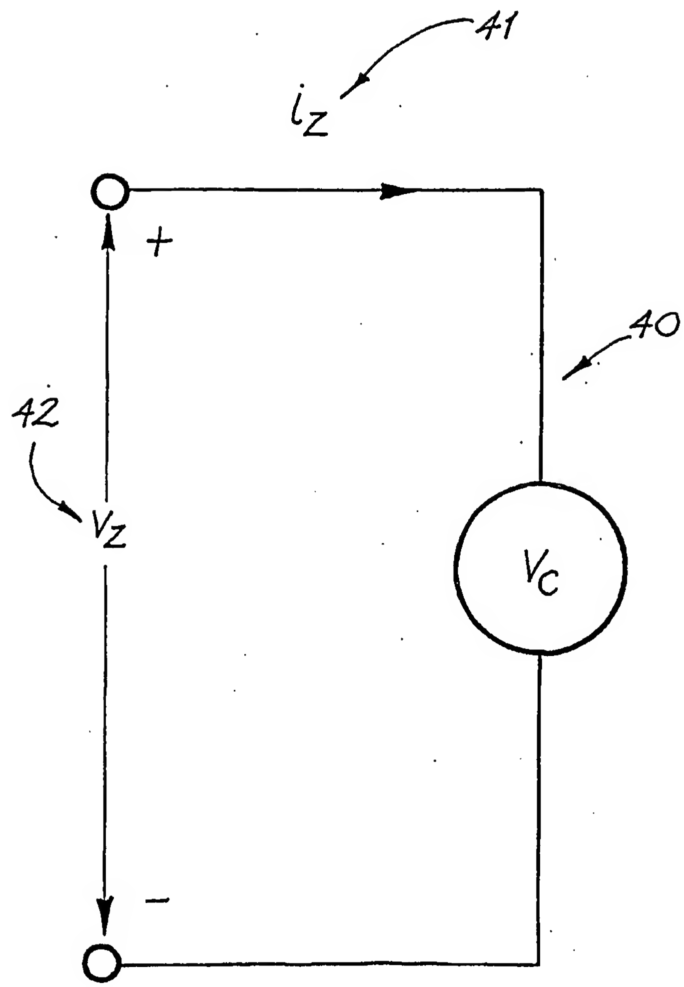

FIG. 9

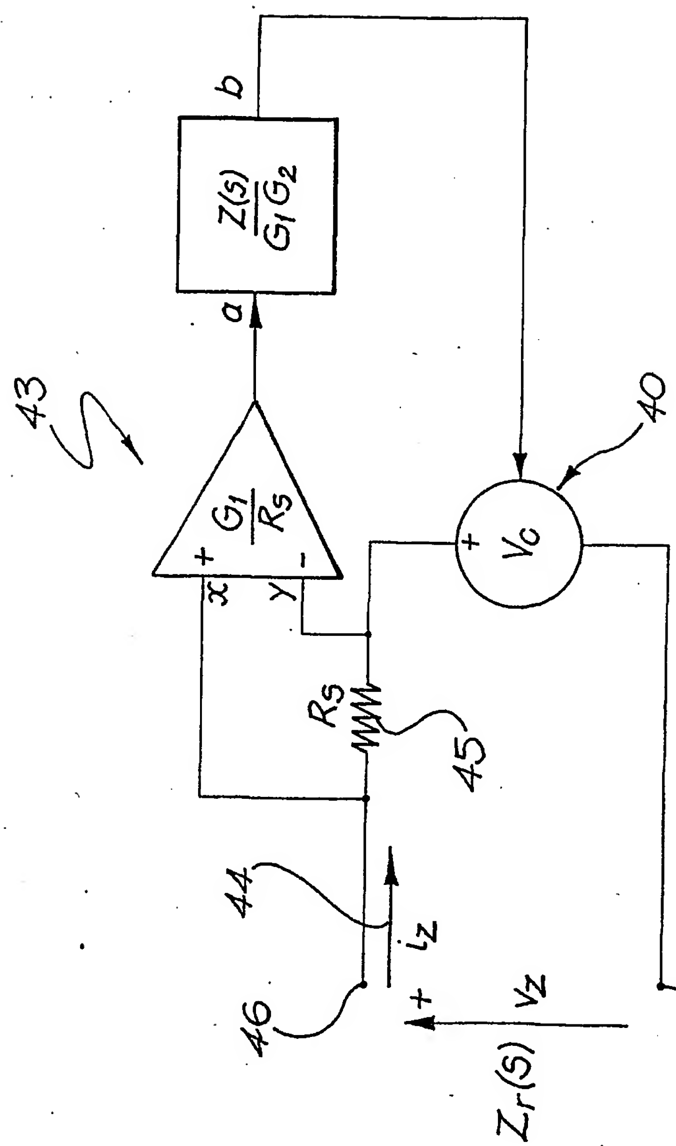

FIG. 10

INTERNATIONAL SEARCH REPORT

International application No. PCT/AU01/00566

A.	CLASSIFICATION OF SUBJECT MATTER
Int. Cl. 7:	H01L 41/107, F16F 15/00

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

WHOLE IPC

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

WHOLE IPC

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

WPAT: Piezo-Electric, Piezo electric, piezoelectric, vibration, structural excitation, damp, control, amplitude reduction, shunt, protect, filter

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
	LEONARD S. BOBROW, "Elementary Linear Circuit Analysis", Holt-Saunders International edition published 1981 by Holt, Rhinehart and Winston, Inc. ISBN 4-8337-0014-X	
X	Pages 6, 7, 32-35, 395, 622, 623, 674-676	1-3
Y	Pages 6, 7, 32-35, 395, 622, 623, 674-676	4,5
	EP 0715092 A2 (AT & T Corp.) 5 June 1996	
Y	Col. 2 line 36-Col. 5 line 13	4,5
	US 5783898 A (Wu) 21 July 1998	
Y	Col. 1 line 61-Col. 2 line 42	4,5

[X] Further documents are listed in the continuation of Box C [X] See patent family annex

* Special categories of cited documents:

"A" document defining the general state of the art which is not considered to be of particular relevance

"E" earlier application or patent but published on or after the international filing date

"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
06 July 2001	12 July 2001
Name and mailing address of the ISA/AU	Authorized officer
AUSTRALIAN PATENT OFFICE PO BOX 200, WODEN ACT 2606, AUSTRALIA E-mail address: pct@ipaustralia.gov.au Facsimile No. (02) 6285 3929	JUZER KHANBHAI Telephone No : (02) 6283 2176

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	WO 99/46858 A1 (THE BOEING COMPANY) 16 September 1999 Fig. 1; Page 4 line 12-Page 7 line 25	4,5

INTERNATIONAL SEARCH REPORT
Information on patent family members

International application No.
PCT/AU01/00566

This Annex lists the known "A" publication level patent family members relating to the patent documents cited in the above-mentioned international search report. The Australian Patent Office is in no way liable for these particulars which are merely given for the purpose of information.

Patent Document Cited in Search Report		Patent Family Member			
EP	715092	JP	8234847	US	5558477
US	5783898	NONE			
WO	9946858	AU	27963/99	US	6075309

END OF ANNEX

(12) INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

CORRECTED VERSION

(19) World Intellectual Property Organization
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(10) International Publication Number
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(25) Filing Language: English

(26) Publication Language: English

(30) Priority Data:
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Published:
— *without international search report and to be republished upon receipt of that report*

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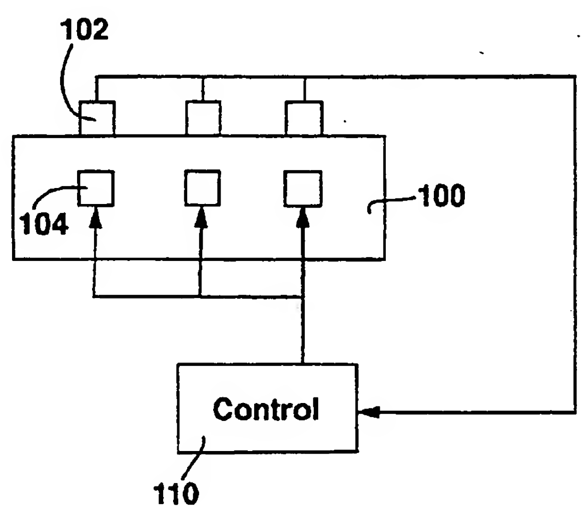
(15) Information about Correction: see PCT Gazette No. 42/2002 of 17 October 2002, Section II

For two-letter codes and other abbreviations, refer to the "Guidance Notes on Codes and Abbreviations" appearing at the beginning of each regular issue of the PCT Gazette.

(54) Title: ACTIVE MANAGEMENT AND STEERING OF STRUCTURAL VIBRATION ENERGY



(57) Abstract: Vibration suppression, isolation, and control in machines, devices, and structures is provided using active vibration control by confinement (AVCC). Vibrational energy is confined to sepecified regions to reduce or amplify the energy. An active vibration energa management system is used to manage vibration in a member. Sensors are coupled to the member to obtain a member response to vibrations. These sensors can be non-contacting or embedded with the member. A signal processor is coupled to the sensors to extract member displacements, temporal derivatives of the displacement and spatial derivatives of displacment. Actuators are coupled to the member to apply feedback forces in response to the signal processor to confine or redirect vibration energy to one or more predetermined member regions.

WO 02/042854 A2

ACTIVE MANAGEMENT AND STEERING OF STRUCTURAL VIBRATION ENERGY

Statement as to Rights Under Federally Sponsored Research and Development

The U.S. Government has a paid-up license in this invention and the right in limited circumstances to require the patent owner to license others on reasonable terms as provided for by the terms of (Contract No. DAAH01-94C-R001) awarded by DARPA (DOD) Defense Small Business Innovation Research Program.

Technical Field of the Invention

The present invention relates generally to vibration and, in particular, the present invention relates to controlling vibration in a system using active vibration confinement.

Background of the Invention

The suppression or control of vibration is having an ever-increasing importance in the design, manufacture, operation, maintenance, precision, and safety of flexible structures and machinery. Engineering systems are subjected to numerous disturbances from either internal or external sources of vibration. The conventional methods for reducing the impact of vibrations take several forms. The methods can be classified, however, into the general categories of isolation, absorption (redirection), and suppression (dissipation). All of these methods can be implemented by passive and/or active (adaptive) means.

Isolation systems reduce the transmission of vibration energy from one part of a structure to another and consist of elastic (isolating) and damping (dissipating) mechanisms optimized for discrete operating frequencies. Examples of the passive isolation elements include, but are not limited to, metal and air springs, elastic mounts, and viscoelastic mounts. These systems are also optimized for some range of operating frequencies to make them most effective.

implemented by passive or semi-active means which controlled the position and/or stiffness of structural or machinery components.

Although inducing mode localization and vibration confinement by design modifications and/or tuning the allowable design parameters in a passive or active way may be useful in many applications, it has potential difficulties in some other applications. The strategy based on design modification is most useful when implemented at the design stage or when allowable structural modifications are sufficient to induce mode localization. Its main drawback, however, is when the allowable design changes fall out of the range for inducing noticeable energy confinement.

For the reasons stated above, and for other reasons stated below which will become apparent to those skilled in the art upon reading and understanding the present specification, there is a need in the art for a vibration confinement system that uses all feedback forces to manage structural elastic energy in order to steer vibration energy and induce confinement.

Summary of the Invention

The above-mentioned problems with vibrating systems and other problems are addressed by the present invention and will be understood by reading and studying the following specification.

The present invention relates to vibration damping, isolation, suppression, absorption, and/or vibration control devices for structures and machinery based on the concept of Vibration Control by Confinement (VCC) in its active (AVCC) form. In one embodiment, sensor(s) and actuator(s) are employed with suitable signal conditioning to apply a feedback force necessary to confine vibration energy to specified regions or components of a structure or machine (hereinafter referred to as "the system"). The term "force" is used to describe any means of inputting mechanical energy into the system (force, moment, strain, pressure). Discrete and/or

distributed sensors and actuators are utilized to modify the system's vibration characteristics to confine or localize vibrational energy.

Contrary to prior techniques of vibration control, the present invention utilizes not only the time-dependent characteristics of the system response, but also their space-dependent characteristics. Through the application of feedback forces proportional to the spatial partial derivatives of the system displacements, velocities, and accelerations, vibration modes are altered to effectively confine, localize, or redistribute the vibration energy in the spatial domain. Furthermore, the vibration response of the system can be suppressed and/or controlled independent of the disturbance type. In conventional methods, the vibration control system must be defined for a specific type of load. In different embodiments, the present invention can be integrated into the design of a system or used as an add-on component. The applications for this invention are far-reaching and can be used in any system whose vibrations at one or more regions must be small or large compared to that of other regions.

The novel characteristics of the current invention are set forth in particular in the appended claims. The invention itself and its possible embodiments are more fully described in the description of the accompanying drawings.

Brief Description of the Drawings

Figure 1 is a block diagram of a system according to the present invention.

Figures 2A-2D are schematic views of four possible embodiments of the present invention in an application where the control system is designed for axial and/or torsional vibration control of a rod or shaft structure.

Figures 3A-3D are schematic views of four possible embodiments of the invention in an application where the control system is designed for axial, torsional, and/or bending vibration of a beam structure.

Figures 4A-4D are schematic views of four possible embodiments of the invention in an application where the control system is designed for in-plane and out-of-plane vibration of a plate structure.

Figures 5A-5C are schematic views of four possible embodiments of the invention in an application where the control system is designed for in-plane and out-of-plane vibration of a shell structure.

Figure 6 is a schematic of one possible embodiment of the invention in an application where the control system is designed for a complex truss structure consisting of beam elements.

Figures 7A-7C are non-dimensional representations of the first three normal vibration mode shapes of a pinned-pinned rod (beam) having no spatial-decay-causing actuators.

Figures 8A-8C are non-dimensional representations of the first three confined vibration mode shapes of a rod (beam) when actuator feedback gains are proportional to the spatial derivatives of displacement according to the present invention.

Figures 9A-9C display the displacements of the non-controlled and controlled rods of Figures 7A-7C and Figures 8A-8C subjected to an impact of 1 m from its left edge at three time increments after impact.

Figure 10 illustrates system energy for a non-controlled and controlled rod as a function of time

Detailed Description of the Invention

In the following detailed description of the preferred embodiments, reference is made to the accompanying drawings that form a part hereof, and in which is shown by way of illustration specific preferred embodiments in which the inventions may be practiced. These embodiments are described in sufficient detail to enable those skilled in the art to practice the invention, and it is to be understood that other embodiments may be utilized and that logical, mechanical and electrical changes

may be made without departing from the spirit and scope of the present invention. The following detailed description is, therefore, not to be taken in a limiting sense, and the scope of the present invention is defined only by the claims.

The present invention uses active feedback actuation to confine vibration energy to specified regions or components of a mechanical system by modifying the system's vibration characteristics, referred to herein as Active Vibration Control by Confinement (AVCC). This approach is distinctly different than prior active vibration control techniques in that this technique utilizes not only the time-dependent characteristics of the system response, but also their *space-dependent* characteristics. Through the application of feedback forces proportional to the *spatial* partial derivatives of the system displacements, velocities, and accelerations, vibration modes are altered to effectively confine or redistribute the vibration energy in the spatial domain. Contrary to conventional methods, the vibration response of the system can be controlled independently of the type of disturbance. Embodiments of the present invention can be both integrated into the design of a system or used as add-on components.

The present invention is differentiated from the prior vibration control methods on several levels. Perhaps the most significant difference lies in the confinement of vibration energy itself. For all prior techniques, the reduction of vibration assumes that the control mechanism responds to the incoming vibration. That is, the systems are reactive. The present approach, on the other hand, prohibits specified regions of a system from accepting vibration energy. In this sense, the approach is proactive.

All currently available methods of vibration control assume that vibration will propagate into a control region. That is, the unwanted vibration is addressed only after the vibration has reached a critical area. Specifically, for isolation techniques, it is assumed that vibration will be present in a system at the interface between two components. It is at this interface that the isolation reacts to the incoming vibration, reducing its propagation. For the case of absorption, without

vibration being delivered to the absorption mechanism, it is ineffective. This control method then reduces the vibration energy in the remaining system components. Suppression techniques are most effective when large amounts of energy are delivered to the damping mechanism. In this case, it is first necessary for vibration to be present in the system, and only then is it dissipated. Prior art active vibration control techniques rely on the application of forces that counteract and cancel the vibration present in the system. For these control mechanisms, it is assumed that vibration will first reach an unwanted region, and then will be canceled. It is clear that each of these mechanisms operates in a reactive mode.

In the instant invention, feedback forces proportional to the spatial derivatives of the system displacements, velocities, and/or accelerations induce confinement. The result is spatial vibration confinement in the form of an exponential decay in vibration magnitude along the length of the structure or its components. As a result, vibration is confined to non-critical regions of the system, preventing vibration energy from propagating to regions of the system that must remain vibration free.

The current invention has significant advantages over the other available methods. The application of AVCC to vibrating systems allows specified regions to reach an acceptable level of vibration faster than prior approaches. It is conceivable that the current invention may be implemented to simply prohibit vibrational energy from propagating into the critical regions. The current invention, however, has the capability to reduce the absolute level of vibration to levels below that of prior techniques. The redistribution of vibration energy, as embodied in the current AVCC invention, dictates that less energy is needed to redirect the vibration than is required to cancel it. The current invention may require less power and fewer actuators than the prior active vibration cancellation methods since the prior methods require vibration cancellation at all parts of the system.

AVCC can be utilized in conjunction with other vibration suppression techniques, active or passive, to increase their effectiveness. If used to amplify

vibration, AVCC can direct more energy to dissipation mechanisms, thus simulating a resonance condition, and making damping devices more effective. Another application of the present invention is in the area of active sonars in which it is desired to transmit signals with the least input power.

Examples of systems which may benefit from this method of vibration control include, but are not limited to: high-precision pointing, imaging, and cutting devices, computer disks, read-write heads, rotating and reciprocating machinery, civil structures, transportation vehicles, and flexible space structures, submarines, ships, aircraft, and off-shore oil platforms.

Overview of AVCC

In this section, the theoretical aspects of the invention are briefly discussed. AVCC is based on two powerful concepts: (1) a modified version of the well-known independent modal-space control (IMSC) approach and (2) the theory of mode localization, whose occurrence in a structure is controlled by a set of actuators rather than design modifications. We refer to such actuators as Spatial-Decay-Causing Actuators (SDCA). A set of discrete (or distributed) feedback forces are used to maintain confinement and/or assist in converting the original extended modal response of the structure into exponentially decaying functions in the spatial domain, while removing some of the energy. These processes result in the simultaneous decay of vibrations in time and spatial domains. In most of today's practices in passive and active vibration control, the total response only decays in the time domain. Therefore, the novel AVCC technique significantly improves the effectiveness and efficiency of vibration control systems by utilizing an optimally distributed sensor/actuator set.

Structures can be designed with the aid of the AVCC technique to optimally distribute the aerodynamically, hydrodynamically, and mechanically induced vibrational energy away from certain critical regions through which transmission of the vibrational forces to the damageable subsystems is maximum. Such an approach

results in a significant improvement in the vibration attenuation of structures during their normal use and thereby optimizes their service-free life and enhances their performance.

The basic process, that takes place when vibration confinement is implemented and maintained via an array of Spatially-Decay-Causing Actuators (SDCA), is that the excess energy is taken from the critical areas (or sensitive components) and placed in the non-critical areas (or non-sensitive components). In doing so, some of the energy is also taken from the structure. Another way of looking at the AVCC approach via SDCA is that certain conditions are provided in the structure so that the vibrational energy is redirected from one place to another and from one mode to another.

The concept for the confinement of vibrational energy described in this section is closely related to that of Independent Modal Space Control (IMSC) or eigenstructure assignment that has been known for some time. The IMSC method has been widely used in the control of linear time invariant systems. Although the proposed method for vibration confinement uses the concept of IMSC (or eigenstructure assignment), an innovative scheme is presented for shaping the vibration modes and/or shifting frequencies using appropriate feedback forces.

Modal control means controlling the vibrations of a structure by controlling its modes or eigenstructure (eigenvalues and eigenvectors). It is well known, based on linear vibration theory, that the transient vibrations of a structure depend on its (1) eigenvalues, which determine the decay/growth rate of the response, (2) eigenvectors, which determine the shape (or the relative distribution) of the response, and (3) initial conditions, which determine the degree that each mode participates in the transient response. For steady-state vibrations (due to persistent disturbances), the magnitude and shape of the response depend on eigenvalues, eigenvectors, and the persistent disturbance whose distribution and frequency determines the degree of participation of each mode in the steady-state response. Thus, system response (transient and steady-state) can be altered by modifying the

eigenstructure (eigenvalues and eigenvectors) of the structure by appropriate feedback forces. It is important to note that all the reported control strategies based on the IMSC method lead to time-decay of vibrations at a certain rate throughout the structure. In other words, vibrations decay as a function of time and not of the spatial parameters.

Active Vibration Control by Confinement (AVCC)

The current invention describes the utilization of feedback forces proportional to the spatial derivatives of system displacements, velocities, and/or accelerations to control the distribution of vibration energy in a structure or machine. When applied in the proper proportions, these feedback forces have the capacity to produce an exponentially varying vibration response magnitude in a structure or component. As such, the system's response may be tailored either to suppress or amplify vibration at specified regions or components. Embodiments of the present invention include sensors, signal processing, and actuators to monitor the response of the structure, calculate the spatial derivatives of the system displacements, velocities, and accelerations, and apply the necessary feedback forces.

The invention may be applied to structurally simple components such as rods, beams, plates, and shells, or to more complex structures comprised of the former types of components. The invention can be utilized also for machines that undergo vibratory motion due to their own operation. The invention may be implemented either in the design stage of a system, or as an add-on component.

Figure 1 illustrates a block diagram of an embodiment of the present invention. The embodiment encompasses embodiments described below with reference to Figures 2A-5. The invention utilizes a device for measuring the vibration response of a system 100. Depending upon the system under consideration, the device(s) 102 may measure displacement, velocity, acceleration, pressure, and/or strain. They may also be contacting or non-contacting sensors. From the vibration measurement, the distribution of displacements, velocities, and

accelerations is derived through signal processing 110. The spatial partial derivatives of these system quantities are then computed. Signals proportional to these computed quantities, called the feedback gains, are then fed to an array of system actuators 104. The feedback gains for the spatial derivatives of the displacements control the rate of spatial decay and the severity of confinement. The feedback gains for the spatial derivatives of velocity create simultaneous spatial and time decay. The feedback gains for the spatial derivatives of the accelerations adjust the shift in system natural frequencies. By describing a consistent set of feedback gains and forces, it is possible to shift the confinement (or suppression) region to a specified location. The application of the feedback forces then alters the vibration mode shapes of the system.

It has been seen in the case of passive vibration confinement that should confinement occur naturally, but unexpectedly, vibration control systems may prove to be ineffective. The primary reason for this result is that the vibration modes of the system are different from those that are expected. Prior active vibration control methods utilize an optimum distribution of sensors and actuators. Once the vibration modes of the system are altered, the effectiveness of the sensors, actuators, and control algorithm may be jeopardized. In contrast, with the application of AVCC, the optimum feedback forces for all actuators are determined by the current state of the system. As such, vibration confinement will be maintained and the necessary vibration suppression or amplification will be ensured.

Figures 2A-2D illustrate four possible embodiments of the invention as it could be applied to the confinement of axial and/or torsional vibrations of a rod 150. In these configurations, Figure 2A and Figure 2B represent, respectively, embodiments with distributed (continuous) sensor and/or actuator array in a surface mount configuration 120 and an embedded configuration 130. Figure 2C and Figure 2D represent the same rod with discrete sensor and/or actuator array in a surface mount configuration 140 and an embedded configuration 160, respectively. The sensors and actuators may be located at the same points on the rod (collocated) or at

different points (non-collocated). Although a solid rod of circular cross-section is illustrated, the cross-section of the rod may take any geometry and may also be hollow. The graphic representation of the rod is used also to identify the possibility of implementing the current invention for the case of rotating components, such as a rotating shaft, as well as non-rotating components. A controller 110 handles all data acquisition functions, signal conditioning and processing, and actuator signal generation.

Figures 3A-3D illustrate four possible embodiments of the invention as it could be applied to the confinement of axial, torsional, and/or transverse vibrations of a beam 170. In these configurations, Figure 3A and Figure 3B represent embodiments with distributed (continuous) sensor and/or actuator array in a surface mount configuration 200 and an embedded configuration 210, respectively. Figure 3C and Figure 2D illustrate the same beam with a discrete array of sensors and/or actuators in a surface mount configuration 220 and an embedded configuration 230. The sensors and actuators may be located at the same points on the beam (collocated) or at different points (non-collocated). Although a solid beam of rectangular cross-section is depicted in the plots, the cross-section of the beam may take any geometry and may also be hollow. The current invention may be utilized for the case of rotating beam-type components as well as non-rotating components. The controller 10 handles all data acquisition functions, signal conditioning and processing, and actuator signal generation.

Figures 4A-4D illustrate four possible embodiments of the invention as it could be applied to the confinement of in-plane and out-of-plane vibrations of a plate 250. In these configurations, Figure 4A and Figure 4B represent embodiments with distributed (continuous) array of sensors and/or actuators in a surface mount configuration 260 and an embedded configuration 220, respectively. Figure 4C and Figure 2D illustrate the same plate to be controlled with a discrete array of sensors and/or actuators in a surface mount configuration 280 and an embedded configuration 290. The sensors and actuators may be located at the same points on

the plate (collocated) or at different points (non-collocated). The current invention may be utilized for the case of rotating plate-type components as well as non-rotating components. All data acquisition functions, signal conditioning and processing, and actuator signal generation are handled by the controller 110.

Figure 5A-5C illustrate three possible embodiments of the invention as it could be applied to the confinement of axial, torsional, bending, and circumferential vibrations of a shell 300. In these configurations, Figure 5A and Figure 5B illustrate a distributed (continuous) array of sensors and/or actuators in a surface mount configuration 310 and an embedded configuration 320. Figure 5C illustrates the same shell to be controlled with a discrete array of sensors and/or actuators in a surface mount configuration 330. The sensors and actuators may also be embedded configuration for application to shell-type components. The sensors and actuators may be located at the same points on the shell (collocated) or at different points (non-collocated). The current invention may be utilized for the case of rotating shell-type components as well as non-rotating components. The controller 110 handles all data acquisition functions, signal conditioning and processing, and actuator signal generation.

Figure 6 illustrates one possible embodiment of the invention as it could be applied to the confinement of vibrations of a complex structure 350. In the depicted configuration, the structure is comprised of beam- and plate-type components to form a complex truss structure. Any complex structure is typically comprised of the aforementioned components (rods, beams, plate, and shells) and the current invention can be utilized for the confinement of vibration in all complex structures. Figure 6 represents the case of a discrete array of sensors and actuators 360 in a surface mount configuration. The discrete sensors and actuators may also be embedded within the complex system. Distributed surface mount and embedded sensors and/or actuators may also be utilized. The sensors and actuators may be located at the same points on the structure (collocated) or at different points (non-collocated). The current invention may be utilized for the case of rotating complex

structures as well as non-rotating structures. The controller 110 handles all data acquisition functions, signal conditioning and processing, and actuator signal generation.

It is understood that each of the devices and elements or a combination of them described herein may also be applied to other structures and machine application where vibration needs to be suppressed, isolated, absorbed, amplified, or controlled by the AVCC concept described above. In particular, AVCC can be applied to flexible structures and/or machinery whose gross motion is either static or dynamic (translating and/or rotating).

An algorithm used by the controller is described herein for implementation of AVCC in which the reshaping of modal response plays the key role while the frequency shift plays a secondary role. In this approach, the mode shapes can be used to effectively distribute the vibrational energy of flexible structures so that the vibration control system can more efficiently meet the required performance. In particular, feedback forces are chosen to control the shape of the modes in a way that the relative magnitude at certain regions remain small (or large) for all times. Such localization of vibrational energy can be used to either isolate parts of the system from some other ones (i.e., vibration isolation) or amplify the output of the structure at certain regions (i.e., amplification of the actuator output displacement).

Example 1

In the following section, the present AVCC method is illustrated for two cases of a continuous structure with collocated distributed sensors and actuators, and a continuous structure with discrete sensors and actuators.

The dynamic response of many non-gyroscopic engineering systems is governed by equation (1) which relates the displacement $u(x,t)$ at the equilibrium position of a structure defined in compact domain D and subjected to the actuating and disturbance force distribution $f_a(x,t)$ and $f_d(x,t)$, respectively. $M(x)$ is a positive function describing the mass density, ξ is the damping coefficient, and L is a linear

time-invariant, symmetric, non-negative differential operator representing the stiffness distribution of the structure. x is the spatial variable in domain D. $B[u(x,t)]$ is a set of linear differential operators characterizing the boundary conditions.

$$M(x)\,\ddot{u}(x,t) + 2\xi\left[M(x)L\right]^{1/2}\,\dot{u}(x,t) + Lu(x,t) = f_a(x,t) + f_d(x,t)\ ,\quad B\left[u(x,t)\right] = 0 \quad (1)$$

The distributed feedback force, $f_a(x,t)$, depends on the displacement $u(x,t)$, velocity, and acceleration fields, and their spatial partial derivatives. An alternative combination, which has been amply used and demonstrated in today's common practices in the field of active control, includes displacement, velocity, and acceleration fields. However, little attention has been given to the case when the feedback force consists of spatial partial derivatives of the displacement, velocity, and acceleration. In fact, the proper selection of the distributed force, $f_a(x,t)$, which includes spatial partial derivatives of the displacement, is the essential tool for the vibration confinement based on AVCC via SDCA approach.

In this illustrative formulation, a collection of the displacement and acceleration fields, and their spatial partial derivatives, form an effective actuator feedback force expressed by equation (2).

$$f_a(x,t) = L_a\left[u(x,t)\right] + M_a\left[\frac{\partial^2 u(x,t)}{\partial t^2}\right] \qquad (2)$$

where L_a and M_a are linear spatial differential operators represented by equations (3) and (4), for example. Such a selection of the feedback force is illustrated for the following cases in which α and β are constants that control the rate, extent, and severity of confinement. For the remaining description of the invention, α and β are referred to as proportionality constants or feedback gains.

Flexural vibrations of a beam:

$$L=\left(\frac{EI}{\rho A}\right)\frac{\partial^4}{\partial x^4}\ ,\ M\ =1$$
$$L_a=\alpha_3\frac{\partial^3}{\partial x^3}+\alpha_2\frac{\partial^2}{\partial x^2}+\alpha_1\frac{\partial}{\partial x}+\alpha_0\ ,\ M_a=\beta_0 \qquad (3)$$

In-plane vibrations of a membrane:

$$L=\left(\frac{EA}{\rho A}\right)\left(\frac{\partial^2}{\partial x^2}+\frac{\partial^2}{\partial y^2}\right),\ M=-1$$
$$L_a=\alpha_{1x}\frac{\partial}{\partial x}+\alpha_{1y}\frac{\partial}{\partial y}+\alpha_0\ ,\ M_a=\beta_0 \qquad (4)$$

As it was pointed out, this approach, for confining vibrations in flexible structures, consists of converting the extended mode shapes into exponentially decaying functions of the spatial coordinates by an appropriate selection of the feedback force, $f_a(x,t)$, in equations (1) and (2). Furthermore, it is possible to select the feedback forces so that the spatial confinement of vibrational energy works in tandem with time decaying functions as it is shown in the following general form.

$$f_a(x,t)\ =\ L_a[u(x,t)]\ +\ C_a\left[\frac{\partial u(x,t)}{\partial t}\right]\ +\ M_a\left[\frac{\partial^2 u(x,t)}{\partial t^2}\right] \qquad (5)$$

It is possible to design the actuating force by utilizing the classical pole placement methods, such as using the Routh-Hurwitz criterion or the Lyapnov stability technique, and applying them to the spatial domain. In particular, the proportionality constant coefficients in equations (3) and (4), for example, must be properly selected in order to stabilize the spatial dynamics. The concept of spatial domain stability is introduced as a concept similar to that of the time domain stability.

To further clarify the Active Vibration Control by Confinement (AVCC) technique via SDCA, the concept is applied to the case of axial vibrations of a rod's normal modes (extended or non-localized) that are converted to exponentially decaying mode shapes. Either distributed or discrete feedback forces can achieve vibration confinement.

The equation of motion of a uniform rod with fixed ends is given in equation (6). Here E is the Young's modulus, A is the cross-sectional area, $u(x,t)$ is the axial displacement, L is the length, ρ is the mass density of the rod, and $f_a(x,t)$ is a possible feedback force applied to the rod through a distributed PZT actuator, for example.

$$-\frac{\partial^2 u(x,t)}{\partial t^2}+\left(\frac{EA}{\rho A}\right)\frac{\partial^2 u(x,t)}{\partial x^2}=\left(\frac{1}{\rho A}\right)\left(f_a(x,t)+f_d(x,t)\right)\ ,\quad u(0,t)=u(L,t)=0 \quad (6)$$

First, to determine the natural frequencies and mode shapes of the rod, consider the case when the feedback and disturbance forces are zero. The frequencies and mode shapes are represented by the following equation.

$$-\frac{\rho A\,\partial^2 u(x,t)}{\partial t^2}+\frac{EA\,\partial^2 u(x,t)}{\partial x^2}=0\ ,\quad \omega_n=\frac{n\pi}{L}\sqrt{\frac{E}{\rho}}\ ,\quad U_n(x)=\eta_n\sin\left(\frac{n\pi x}{L}\right)\ ,\quad n=1,2,3,.. \quad (7)$$

Note that the normal modes, $U_n(x)$, are simple sine waves extended along the entire length of the rod (i.e., the modes are not localized). Currently in the field of active vibration control, it is assumed that natural modes of the rod, for example, are the extended sine waves (see the set of mode shapes displayed in Figures 7A-7C). It is suggested that if the modes are exponentially decayed, the active vibration control would be much more effective.

Second, consider how localizing the mode shapes via the feedback force can exponentially confine the vibrational energy. For example, the case when the

feedback force is proportional to slope and displacement and the disturbance force is zero, as given by (8).

$$f_a(x,t) = -2EA\left(\alpha\frac{\partial u}{\partial x} + \alpha^2 u\right),\ f_d(x,t) = 0 \tag{8}$$

Substituting (8) into (6), grouping the similar terms, and solving for the characteristic equation yields equation (9).

$$\lambda^2 + (2\alpha)\lambda + \left(\alpha^2 + \frac{\rho}{E}\omega^2\right) = 0 \Rightarrow \lambda = -\alpha \pm j\sqrt{\frac{\rho}{E}}\omega\ ,\ j = \sqrt{-1} \tag{9}$$

Assuming that the controllable parameter, α, is a positive value and applying the zero boundary conditions, equation (9) leads to the unchanged natural frequencies and exponentially decaying mode shapes described by (10).

$$\omega_n = \frac{n\pi}{L}\sqrt{\frac{E}{\rho}}\ ,\ U_n(x) = \eta_n\, e^{-\alpha x}\sin\left(\frac{n\pi x}{L}\right)\ ,\ n = 1,2,3,\ldots \tag{10}$$

In this case, note that the natural frequencies of the original rod (uncontrolled rod) are preserved, but the mode shapes are altered for the purpose of vibration confinement.

Figures 7A-7C show non-dimensional mode shapes of the non-controlled, and represent the non-confined normal vibration modes of the rod/beam with no feedback forces applied. Notice that the displacements for the first normal mode 400 in Figure 7A and the third normal mode 404 in Figure 7C are symmetric about their centers 406 and 410, and the displacement of the second normal mode 402 in Figure 7B is skew-symmetric about its center 408. In all cases, the elastic energy

stored in both halves of the rod are equal. Note that the rod/beam length has been normalized to one.

Figures 8A-8C illustrate mode shapes of a controlled (confined) rod when the controlling parameter α equals 0.5. The size of the confined region is determined by the spatial decay rate, α. Also, note that the confinement illustrated in Figures 8A-8C is near the left edge (x=0) of the rod and thus it is expected that the vibration response of the rod subjected to an external excitation would be confined in the same region for all times.

Thus, fundamental results of one embodiment of the present invention are depicted in Figure 8A-8C. These plots represent the normalized axial (transverse) vibration mode shapes of a rod (beam) type.structure. The data was derived in closed form for the case of a rod pinned at its two ends. Figures 8A-8C illustrate the confinement of vibration energy once the feedback forces are applied. In this case, the feedback forces have been set in order to confine the vibration energy to the left region of the rod. Figure 8A, Figure 8B, and Figure 8C represent the first 420, second 422, and fourth 424 confined modes of the rod, respectively. For this example, no feedback forces proportional to the spatial derivatives of acceleration are included. As a result, the natural frequencies of the non-confined and confined modes are unchanged. It is clear from the figures that in all three modes, (1) the vibration energy has been redistributed to the left side of the rod, (2) the energy is confined to the region near the left end of the rod, and (3) that the vibration on the right hand side of the beam has been suppressed.

Based on this illustrative example, the following should be noted:

1. The natural frequencies of the rod can also be altered by adding acceleration terms to the feedback force.
2. If one wishes to confine vibrations near an arbitrary point instead of the point x=0, the origin should be first shifted, and then the feedback control force must be altered.

3. Vibrations can also be confined about N points ($N \geq 1$) by employing a feedback force with (2N-1) discontinuity and 2(2N-1) continuity conditions.
4. The feedback force given by equation (8) has no dissipative term. The force can be modified to include energy dissipation terms, which result in simultaneous confinement and suppression of vibrational energy.

Example 2

In the following section, the distributed feedback force is replaced by a finite set of discrete forces, more practical for geometrically complex surfaces. Equation (11) represents the feedback force consisting of N_a discrete forces.

$$f_a(x,t) = \sum_{j=1}^{N_a} F_j(t)\delta(x-x_j) + r(x,t) \tag{11}$$

$F_i(t)$ is the jth feedback force due to the jth actuator at the location x_i. $r(x,t)$ is a residual function characterizing the error resulting from the replacement of the distributed feedback force by the discrete feedback forces. To determine $F_i(t)$ and $r(x,t)$, one should combine equations (2) and (11), as shown below.

$$f_a(x,t) = L_a[u(x,t)] + M_a\left[\frac{\partial^2 u(x,t)}{\partial t^2}\right] = \sum_{j=1}^{N_a} F_j(t)\delta(x-x_j) + r(x,t) \tag{12}$$

Substituting the expansion theorem (13) into (12) yields equation (14). Note that it is realistic to truncate the series solution (13) to a finite number of modes, N. As with all active vibration control methods, active control of all modes is not practical.

$$u(x,t) = \sum_{n=1}^{N} \eta_n(t)U_n(x) \tag{13}$$

$$\sum_{n=1}^{N}\left[\eta_n(t)\, L_a\left[U_n(x)\right] + \ddot{\eta}_n(t)\, M_a\left[U_n(x)\right]\right] = \sum_{j=1}^{N_a} F_j(t)\delta\left(x - x_j\right) + r(x,t) \quad (14)$$

Pre-multiplying both sides of equation (14) by $G(x)M[U_m(x)]$ and integrating over the spatial domain, one obtains equation (15).

$$\sum_{n=1}^{N} K_{mn}\eta_n(t) + \sum_{n=1}^{N} M_{mn}\ddot{\eta}_n(t) = \sum_{j=1}^{N_a} A_{mj}F_j(t)\delta\left(x - x_j\right) + R_m(x,t) \qquad (15)$$

where $G(x)$ is a weighting function that needs to be selected for the structure and $U_m(x)$ is the mth mode shape of the structure. The time independent coefficients and the time dependent $R(t)$ are given below. Equation (15) can be written in the matrix form as shown below.

$$\left[U_m(x)\right] L_a\left[U_n(x)\right]dx\,,\ M_{mn} = \int G(x)\, M\left[U_m\right. \qquad (16)$$

$$\left.(x)\right]\, r(x,t)dx\,,\ A_{mj} = G\left(x_j\right) M\left[U_m\left(x_j\right)^x\right],\ j = 1,2,..N_a \qquad (17)$$

$$[A]F + [R] = [K]\eta + [M]\ddot{\eta}\ ,\ [A]^T = \left[A_c\ ,\ A_r\right]^T$$

$$[R]^T = \left[R_c\ ,\ R_r\right]^T\ ,\ [K] = \begin{bmatrix} K_{cc} & K_{cr} \\ K_{rc} & K_{rr} \end{bmatrix},\ [M] = \begin{bmatrix} M_{cc} & M_{cr} \\ M_{rc} & M_{rr} \end{bmatrix} \qquad (18)$$

where subscripts c and r indicate the control and the residual modes. $[A]$ is NxN_a, $[K]$ and $[M]$ are NxN, F is a N_ax1 vector, and $[R]$ and η are $Nx1$ vectors. Assuming that η can be obtained by employing a modal observer, such as a modal filter, the discrete actuator force vector, F, can be designed according to equation (19), and the

- quantitative measures of the residual vectors can also be determined based on the same equation shown below.

$$F = [A_c]^{-1}[[K_{cc}]\eta_c + [M_{cc}]\ddot{\eta}_c] \;,\; R_c = [K_{cr}]\eta_r + [M_{cr}]\ddot{\eta}_r$$
$$R_r = -[A_r]F + [K_{rc}]\eta_c + [M_{rc}]\ddot{\eta}_c + [K_{rr}]\eta_r + [M_{rr}]\ddot{\eta}_r \qquad (19)$$

Note that the existence of the inverse of $[A_c]$ depends on the location of the discrete actuators. For example, locating actuators at nodal points associated with the mode shapes is not allowed, as expected.

Some of the issues that are important for demonstrating the practicality of the present invention are the effect of the confined modes on the actual response of the structure, the required actuating force and control effort (energy), control spillover, and the stability of the control system. As an example, it is assumed that the rod is subjected to an impact at a location along its length. The impact is characterized by the resulting initial velocity, Vo, at the location, Xo, measured from the left (x=0) of the rod, which is pinned at its two ends. In this case, the weighting function $G(x)$ and the mode shapes are given by equation (20). The rod considered in this example has the following characteristics: L=10m, Vo=1m/s, Xo=1m, ρ=2600kg/m^3, E=70Gpa, A=10E-4m^2. In this example, the goal is to confine vibrations to the left of the rod.

$$U_n(x) = \eta_n\, e^{-\alpha x} \sin\left(\frac{n\pi x}{L}\right) \;,\; G(x) = e^{2\alpha x} \qquad (20)$$

Figures 9A-9C illustrate the axial deflection of the non-controlled 500 (dashed lines) and controlled 502 rod right after the impact, when the wave has reached the middle of the rod, and right before the wave reaches the other end. Note that the initial response of the controlled rod (with confined modes) is almost the same as the non-controlled rod (with non-confined modes), as expected, due to the

design of the control. However, the response of the controlled rod is much smaller than that of the non-controlled rod in regions away from the impact ($x>1m$, where vibrations were to be suppressed) 1 as shown in Figures 9B and 9C. In both confined and non-confined cases, the damping factor is 0.1%, which is considered small damping.

Figure 10 shows the total system energy for the non-controlled 510 (dashed line) and controlled 520 rod as a function of time. The total energy of the controlled rod is slightly more than the uncontrolled rod only when the wave is in the localized area 530, and is much less when the wave is in the suppressed area. However, the small amount of the additional energy is certainly within the practical range.

The above-described approach can be extended to structures such as beams, plates, and shells. The rod model, used as the discussed example, has a closed form solution and is simple enough to be presented here. However, in the case of a beam, plate, or cylindrical shell there is no closed form solution, and thus a discrete model should be utilized.

While the above described invention was illustrated as embodied in the context of controlling structural and machinery vibrations by confinement, it is not intended to be limited to the details shown, since various modifications and structural changes may be made without departing from the spirit of the presented invention.

It is an objective of the present invention to provide a method for actively controlling, in an efficient and effective manner, the distribution of vibrational energy throughout a structure or machine. This objective may be applied for the purposes of both vibration suppression and vibration amplification. It is another objective of the described invention to provide a way to adjust the location and degree of vibration confinement or suppression. It is further an objective to provide a way for adjusting the natural frequencies of a structure in order to extend the useful frequency range of both the structure or machine, and the vibration control system. Additionally, it is an objective of the current invention to provide a solution for the

optimal selection and implementation of system sensors and actuators for the monitoring and controlling of normal and confined modes of vibration.

The above-described embodiments utilize sensors to detect vibration conditions. These sensors can are not limited to any particular type of sensor and can include Piezoelectric, Micro-machined element, Laser, and Optical Fiber sensors.

Piezoelectric technology is commonly and widely used in vibration and shock sensors. In particular, they are often used in accelerometers, force sensors, and pressure sensors. Piezoelectric devices can also be used for velocity and displacement sensors.

In recent years, micro-machined sensors have made tremendous advances in terms of technology and cost effectiveness. Combined with capacitive technology, this advance technology provided a low cost, very low frequency range (from DC-frequency). The technology for measuring the mass displacement is the electrical property of capacitance. Placing two metal plates in parallel forms a capacitor, and the capacitive property is proportional to displacement between two plates.

The micro-machined element with capacitive property is usually applied in accelerometer. Capacitive technology is widely used in very low frequency vibration and shock accelerometer. These sensors are powered with a DC supply. The sensor has limited high frequency measuring range. Capacitance sensing has the potential to provide a wider temperature range, without compensation because of the low thermal coefficient of expansion of materials. This technology can provide both digital and analog sensing signal. Mounting method of this sensor is also the same as piezoelectric technology.

Lasers have been used for a long time in manufacturing and in the laboratory for vibration sensing. A laser can provide a non-contact measurement for minute vibration. This technology is not only used for analyzing material deformation, and imaging vibration modes. It is also applied to vibrometers.

For measuring vibration, a laser beam is divided into a reference and signal. The signal beam is directed into a vibrating structure, and a back-reflected beam is combined with the reference beam. The difference between the reference and signal beam changes when the structure moves. This results in the intensity modulation of the recombined beam. Therefore, the frequency (F_d) of intensity modulation corresponding with a surface velocity (v) is given by $F_d = 2v/\text{wavelength}$. To analyze the recombined beam, it is split between two independent detection channels configured so that the two signals obtained are phase shifted, depending on the motion of the structure. Electronic mixing of these signals with a carrier frequency is used to derive a single, frequency shifted Doppler signal, which is then converted to analog voltage, directly proportional to the instantaneous velocity of the moving surface.

In vibration monitoring, optical fiber can be performed as a strain sensor or an accelerometer. A fiber optic accelerometer uses a microbend fiber optic intensity sensor. The microbend displacement sensor measures the force required to accelerate a mass. The mass is mounted on a thin, wide beam and contains corrugations on its lower surface. The base of accelerometer contains mating configurations and a support for the beam. A fiber is clamped between the two sets of corrugations. Accelerations of the base cause a relative motion between the mass and the base, resulting in microbending and intensity modulation of the optical power in the fiber. That corresponds to the acceleration.

A fiber optic strain sensor can be approached in two different ways. In the first method, like a proximity sensor, light is emitted from transmitting fibers to an interested surface and back to receiving fibers. The distance to the surface is related to the amount of light intensity in the receiving fiber. In the second method, a section of the fibers is attached to an interested surface, causing the fiber to have the same movement as the structure. This movement is causing microbending in the fiber, resulting in intensity modulation. That corresponds to the motion of the surface.

The embodiments described above use actuators that can include Piezoelectric, Shape memory alloy, Electromagnetic Fluid and an Electrical Motor. Piezoelectric technology is not only used in sensors but also applied as an actuator in active vibration control. The piezoelectric effect is used when operating as a sensor. For the actuator, the inverse effect is used. Piezoelectric materials can be used to convert electrical energy into mechanical energy and vice versa. Piezoelectric technology is widely used in precise motion (nanoscale) because of its many useful properties such as repeatability in high frequency, wide load range, and no maintenance. Lead zirconate titanate (PZT) based ceramic materials are the most often used. There are some basic designs for PZT actuators; stack design, laminar design, tube design, and bender type design.

In a stack design, the actuator consists of a stack of ceramic disks separated by thin metallic electrodes. Maximum operating voltage is proportional to the thickness of the disks. Stack design actuators can withstand high pressure and have the highest stiffness of all piezoelectric design actuators. Spring preloaded actuators are considered because ceramics cannot withstand large pulling forces. This design can be used for static and dynamic operation.

In a laminar design, the actuator consists of thin ceramic strips. The displacement of these actuators is perpendicular to the direction of polarization and the electric field. The maximum travel is a function of the length of the strips, and the number of parallel strips determines the stiffness and stability of the element.

In a tube design, the actuators operate on the transversal piezoelectric effect. When a voltage is applied between the outer and inner diameter, the tube contracts axially and radially. When the outside electrode of the tube is separated into four segments, different drive voltages lead to bending of one end.

In a bender-type design, the actuators operate similarly to a bimetallic strip in thermostats. When the ceramic is energized, the metal substrate is deflected with a motion proportional to the applied voltage.

Shape memory alloy technology is one of the newest technologies that can be

applied for low frequency active vibration control. Shape Memory Alloys (SMA's) are materials that have an ability to return to their original shapes. When an SMA is below its transformation temperature, it has very low yield strength and can be easily deformed into a new shape (which it will retain). However, when an SMA is heated above its transformation temperature, it will return to the original shape. If the SMA encounters any resistance during this transformation, it can generate extremely large forces. The most common and useful shape memory materials is Nickel-titanium alloy called Nitinol (Nickel Titanium Naval Ordnance Laboratory).

Electromagnetic fluid is also not a real active actuator but can be used for adjusting a parameter of systems to get the desired performance. Electromagnetic fluids are essentially suspensions of micron-sized, magnetic particles in oil. Under a normal condition, an electromagnetic fluid is a free-flowing liquid with similar properties to normal oil. When a magnetic field is applied, this fluid transforms into a near-solid state in milliseconds. The fluid can be quickly returned to its liquid state when the magnetic field is removed. The degree of change in an electromagnetic fluid is proportional to the magnitude of the magnetic field.

The properties of controllable fluid depend on concentration and density of particles, particle size and shape distribution, properties of carrier fluid, additional additives, applied field, and temper. In a valve mode with fixed magnetic poles, the system might be considered for hydraulic controls, dampers and shock absorbers. The fluids flow through a small gap that the magnetic field is applied. In a direct shear mode with a moving pole, the system may be proper for clutches and brakes, dampers, and structural composites. In a squeeze film mode, the system would be suitable for small motion control and high force applications. It also can be configured for axial or rotary operation.

Electrical motor technology is widely used in manufacturing and the laboratory. For active vibration control, electrical motor has been a choice because of its flexibility and variety of products in the market. There are many developments in electrical motor technology to make them more efficient, have lower power

consumption, higher torque, lower cost and variety applications. Many commercial products of electrical motor can be applied for active vibration control such as linear actuator and active isolation fittings.

Conclusion

An active vibration energy management system has been described that can be used to manage vibration in a member. Sensors are coupled to the member to obtain a member response to vibrations. These sensors can be non-contacting, contacting or embedded with the member. A signal processor is coupled to the sensors to extract member displacements, temporal derivatives of the displacement and spatial derivatives of displacement. Actuators are coupled to the member to apply feedback forces in response to the signal processor to confine or redirect vibration energy to one or more predetermined member regions. As such, a member can be actively managed to confine vibrations to a specific are of the member. Further, the vibrations can be managed to protect specific areas of the member.

Although specific embodiments have been illustrated and described herein, it will be appreciated by those of ordinary skill in the art that any arrangement, which is calculated to achieve the same purpose, may be substituted for the specific embodiment shown. This application is intended to cover any adaptations or variations of the present invention. Therefore, it is manifestly intended that this invention be limited only by the claims and the equivalents thereof.

CLAIMS

What is claimed is:

1. An active vibration energy management system comprising:

a member;

sensors coupled to the member to obtain a member response to vibrations;

a signal processor coupled to the sensors to extract member displacements, temporal derivatives of the displacement and spatial derivatives of displacement; and

actuators coupled to the member to apply feedback forces in response to the signal processor to confine or redirect vibration energy to one or more predetermined member regions.

2 The active vibration energy management system of claim 1 wherein the signal processor controls the feedback forces applied by the actuators such that the feedback forces are proportional to the spatial derivatives of the displacement.

3. The active vibration energy management system of claim 1 wherein the signal processor adjusts the member region of confinement by controlling the feedback forces applied by the actuators.

4. The active vibration energy management system of claim 1 wherein the signal processor controls a number of confinement regions of the member by controlling the feedback forces applied by the actuators.

5. The active vibration energy management system of claim 1 wherein the signal processor controls confined-mode natural frequencies of the member by

controlling the feedback forces applied by the actuators proportional to the spatial derivatives of the acceleration.

6. The active vibration energy management system of claim 1 wherein the member comprises a beam or rod.

7. The active vibration energy management system of claim 1 wherein the member comprises a structure shell or skin.

8. The active vibration energy management system of claim 1 wherein the member comprises a substantially plainer structure.

9. The active vibration energy management system of claim 1 wherein the sensors are distributed on a surface of the member.

10. The active vibration energy management system of claim 1 wherein the sensors are distributed throughout the member.

11. The active vibration energy management system of claim 1 wherein the actuators are distributed on a surface of the member.

12. The active vibration energy management system of claim 1 wherein the actuators are distributed throughout the member.

13. An active vibration energy management system comprising:

a member comprising one or more structures selected from a group comprising a rod, a plate, or a shell;

distributed sensors coupled to the member;

a signal processor coupled to the sensors to determine displacements of the member, temporal derivatives of the displacement of the member and spatial derivatives of accelerations of the member; and

distributed actuators coupled to the member to apply feedback forces in response to the signal processor to confine or redirect vibration energy to one or more predetermined member regions, wherein the signal processor controls the feedback forces applied by the actuators such that the feedback forces are proportional to the spatial derivatives of the displacement.

14. The active vibration energy management system of claim 13 wherein the sensors are distributed throughout the member.

15. An active vibration energy management system comprising:

a member comprising one or more structures selected from a group comprising a rod, a plate, or a shell;

distributed sensors coupled to the member;

a signal processor coupled to the sensors to determine displacements of the member, temporal derivatives of the displacement of the member and spatial derivatives of accelerations of the member; and

distributed actuators coupled to the member to apply feedback forces in response to the signal processor to confine or redirect vibration energy to one or more predetermined member regions, wherein the signal processor adjusts the member region of confinement by controlling the feedback forces applied by the actuators.

16. An active vibration energy management system comprising:

a member comprising one or more structures selected from a group comprising a rod, a plate, or a shell;

distributed sensors coupled to the member;

a signal processor coupled to the sensors to determine displacements of the member, temporal derivatives of the displacement of the member and spatial derivatives of accelerations of the member; and

distributed actuators coupled to the member to apply feedback forces in response to the signal processor to confine or redirect vibration energy to one or more predetermined member regions, wherein the signal processor controls the feedback forces applied by the actuators such that the feedback forces are proportional to the spatial derivatives of the displacement.

14. The active vibration energy management system of claim 13 wherein the sensors are distributed throughout the member.

15. An active vibration energy management system comprising:

a member comprising one or more structures selected from a group comprising a rod, a plate, or a shell;

distributed sensors coupled to the member;

a signal processor coupled to the sensors to determine displacements of the member, temporal derivatives of the displacement of the member and spatial derivatives of accelerations of the member; and

distributed actuators coupled to the member to apply feedback forces in response to the signal processor to confine or redirect vibration energy to one or more predetermined member regions, wherein the signal processor adjusts the member region of confinement by controlling the feedback forces applied by the actuators.

16. An active vibration energy management system comprising:

a member comprising one or more structures selected from a group comprising a rod, a plate, or a shell;

distributed sensors coupled to the member;

a signal processor coupled to the sensors to determine displacements of the member, temporal derivatives of the displacement of the member and spatial derivatives of accelerations of the member; and

distributed actuators coupled to the member to apply feedback forces in response to the signal processor to confine or redirect vibration energy to one or more predetermined member regions, wherein the signal processor controls a number of confinement regions of the member by controlling the feedback forces applied by the actuators.

17. A method for suppressing, isolating, absorbing, redirecting, or controlling vibrations in a system comprising:

obtaining a current system response to the vibrations;

determining system displacements and their temporal and spatial derivatives;

applying feedback forces to actuators coupled to the system to confine or redirect vibration energy to one or more predetermined system regions.

18. The method of claim 17 further comprises adjusting a degree of confinement of the system by controlling the feedback forces proportional to the spatial derivatives of the displacement.

19. The method of claim 17 further comprises adjusting the predetermined system regions by controlling the feedback forces.

20. The method of claim 17 further comprises adjusting a number of predetermined system regions by controlling the feedback forces.

21. The method of claim 17 further comprises adjusting confined-mode natural frequencies of the system by controlling the feedback forces proportional to the spatial derivatives of the accelerations.

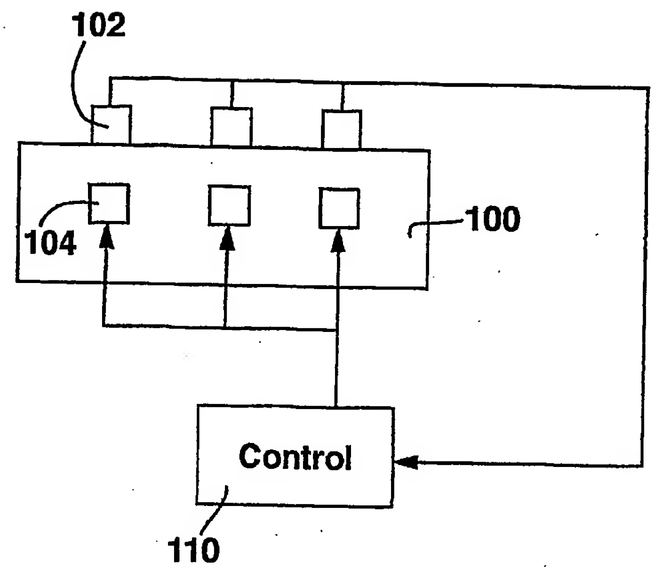


FIG. 1

FIG. 2A

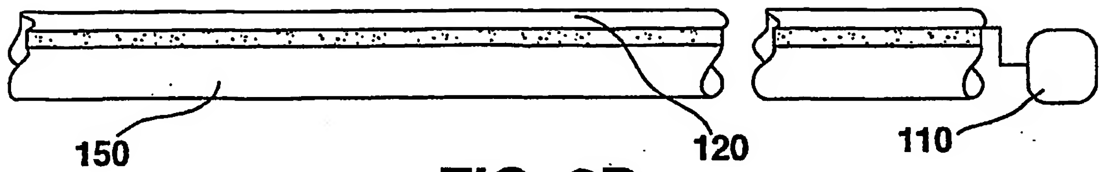


FIG. 2B

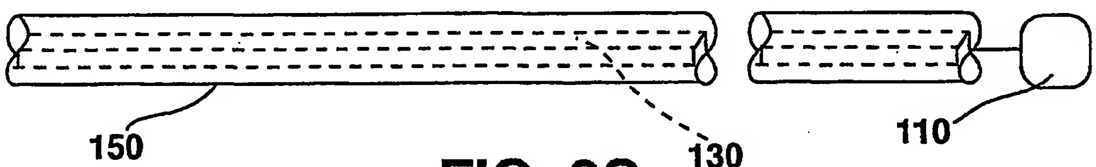


FIG. 2C

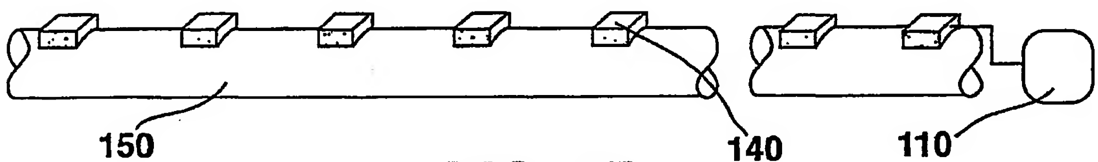


FIG. 2D

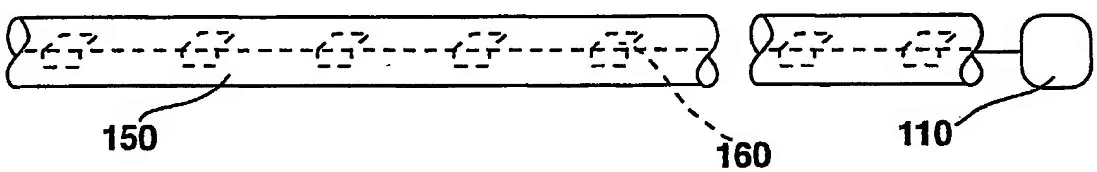


FIG. 3A

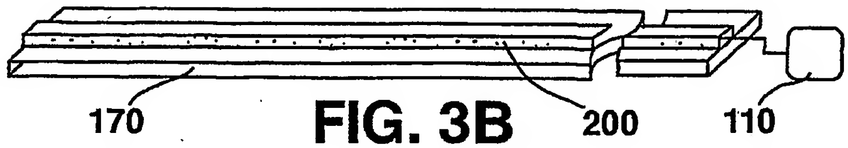


FIG. 3B

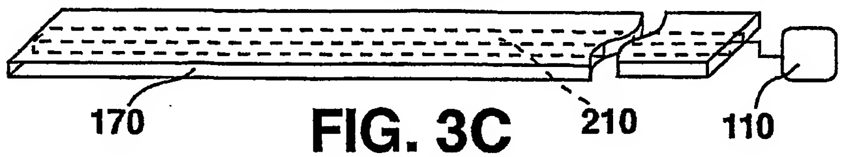


FIG. 3C

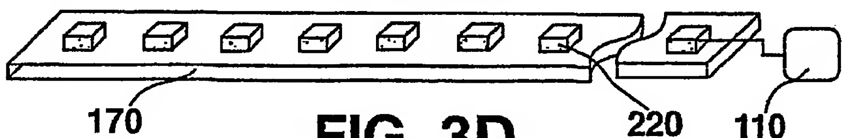


FIG. 3D

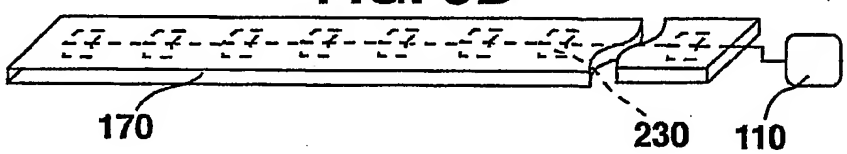


FIG. 4A

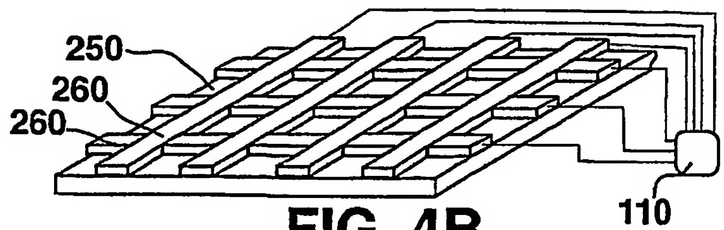


FIG. 4B

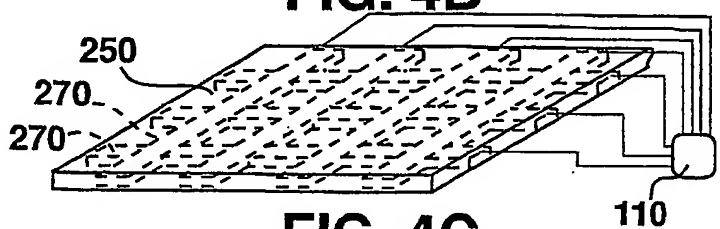


FIG. 4C

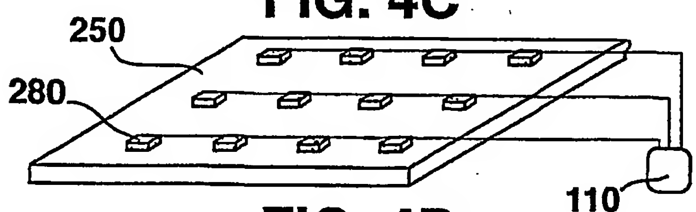


FIG. 4D

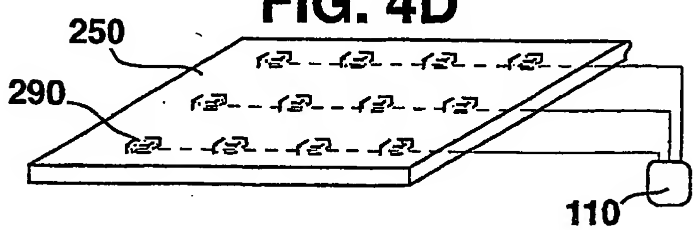


FIG. 5A

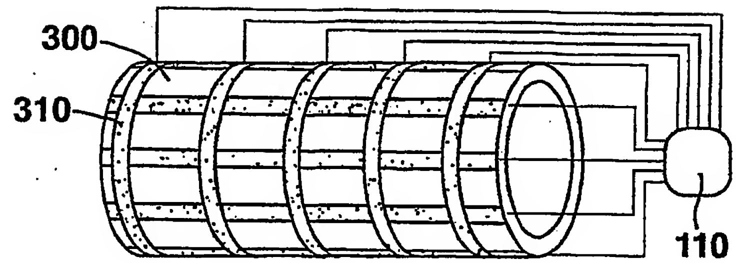


FIG. 5B

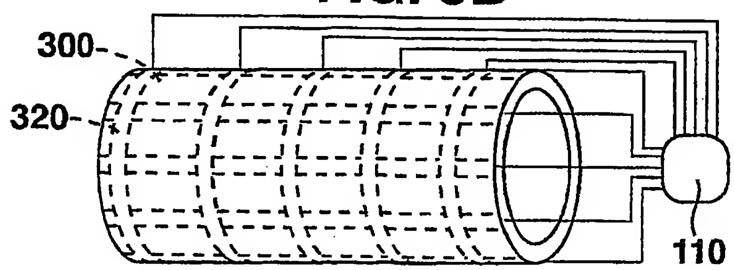
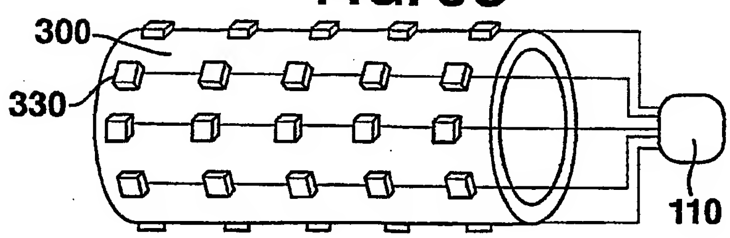


FIG. 5C



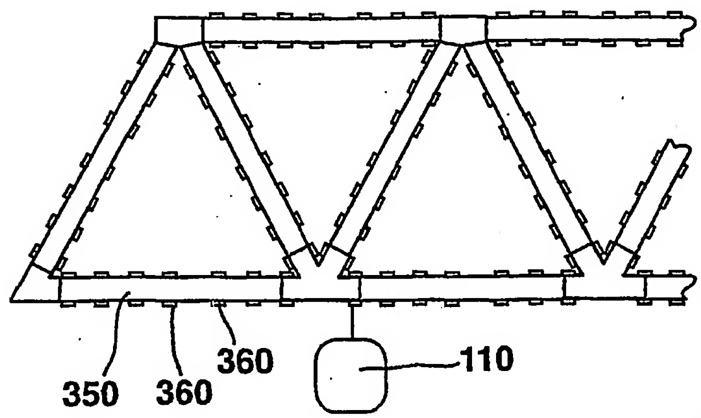


FIG. 6

FIG. 7A

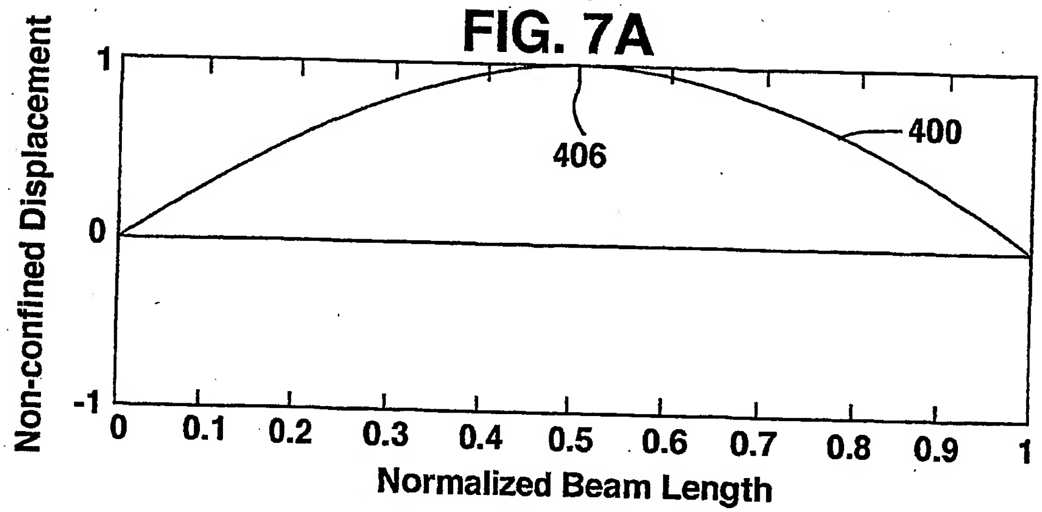


FIG. 7B

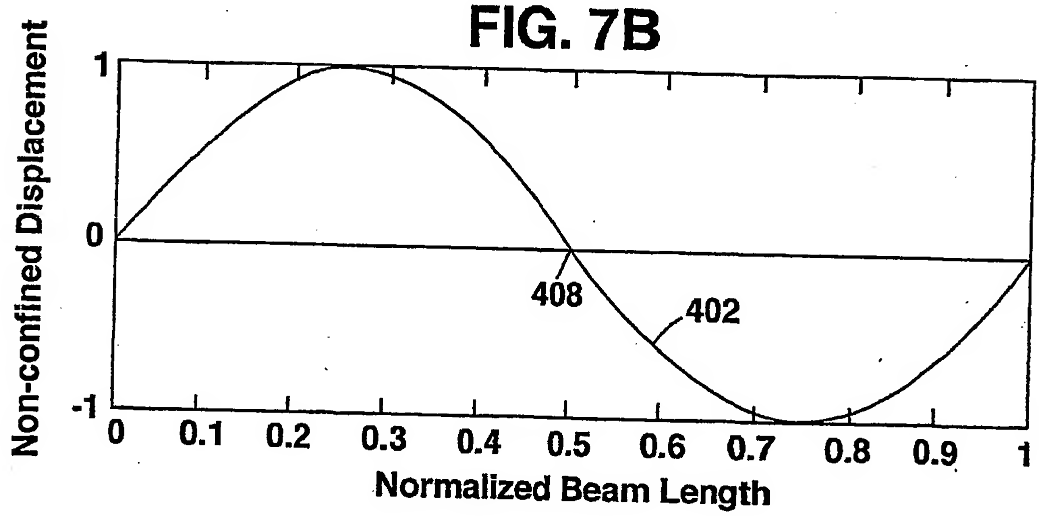


FIG. 7C

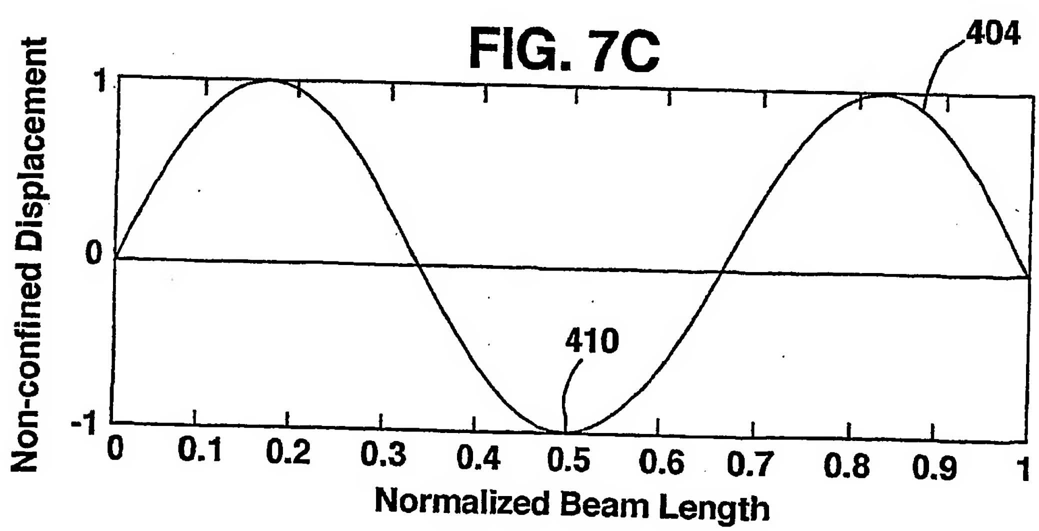


FIG. 8A

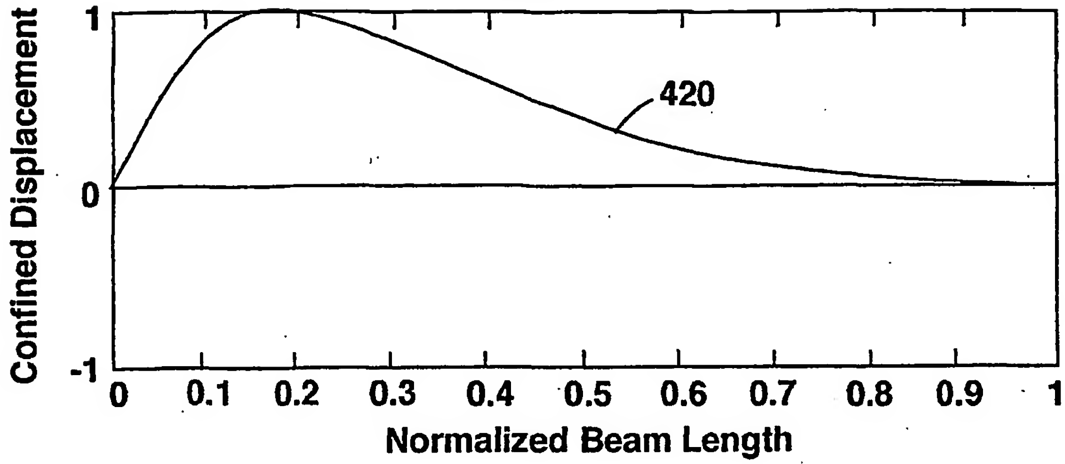


FIG. 8B

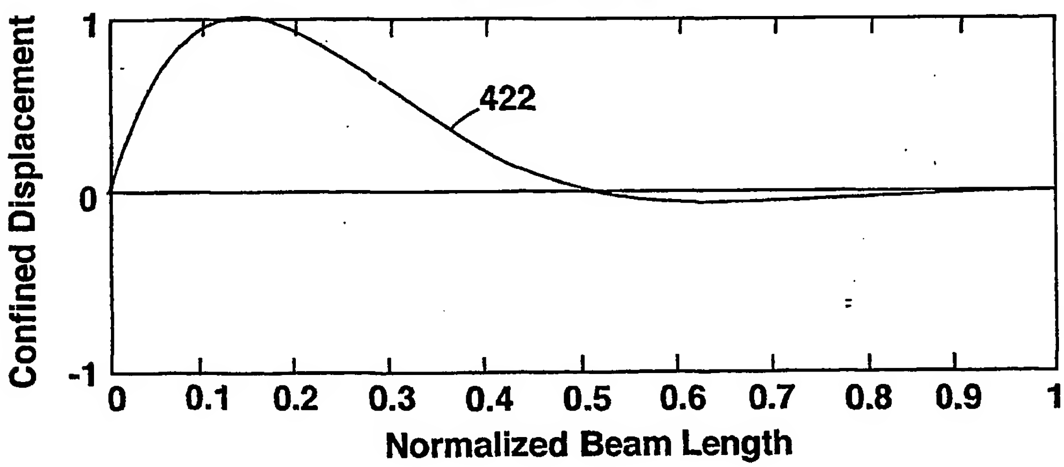


FIG. 8C

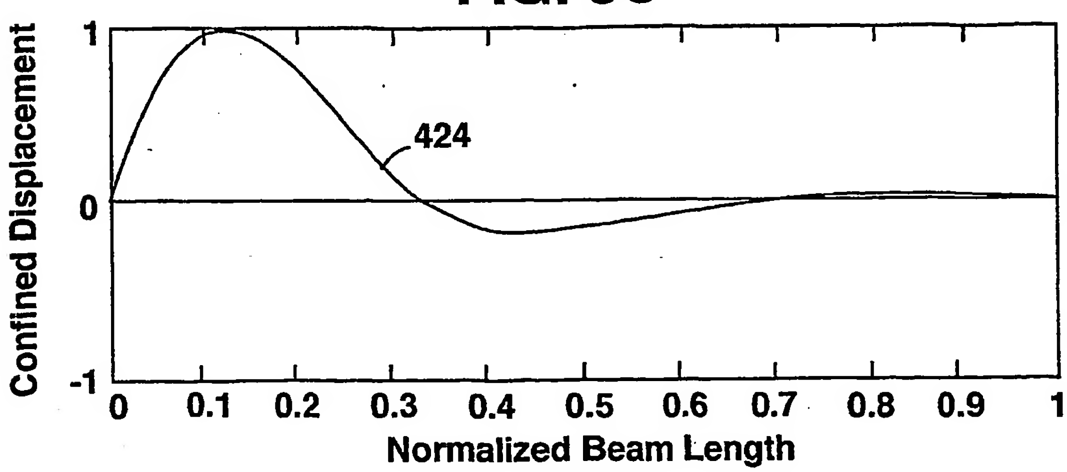


FIG. 9A

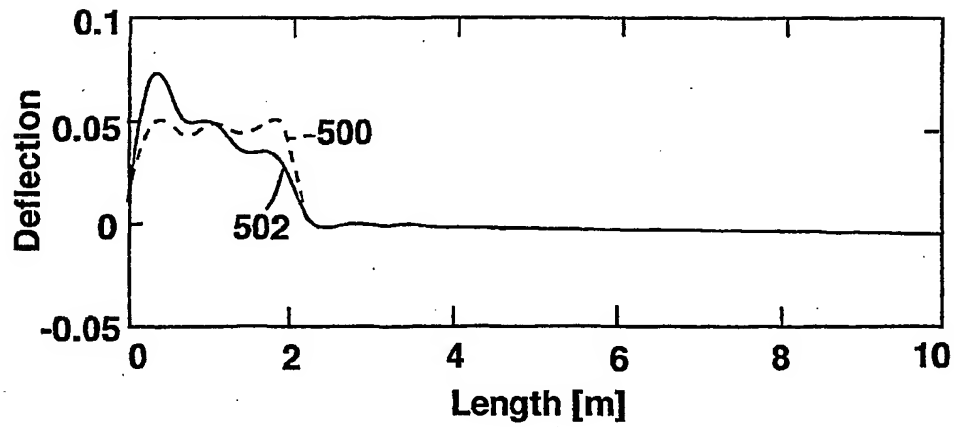


FIG. 9B

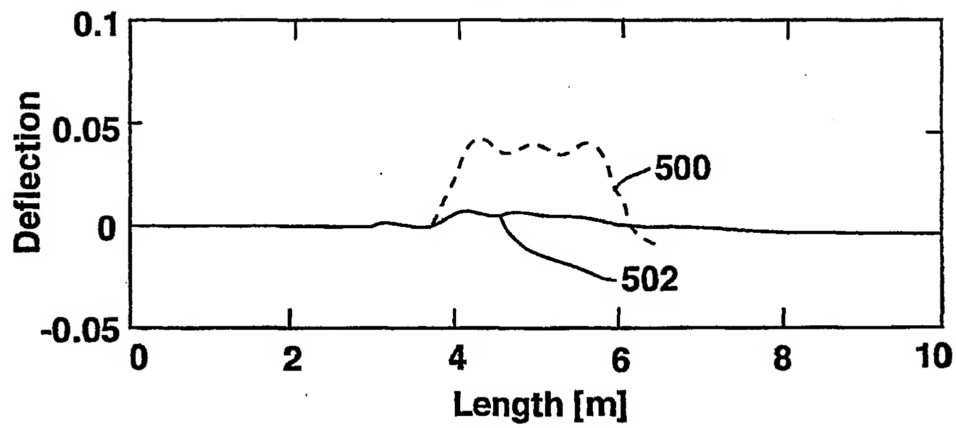


FIG. 9C

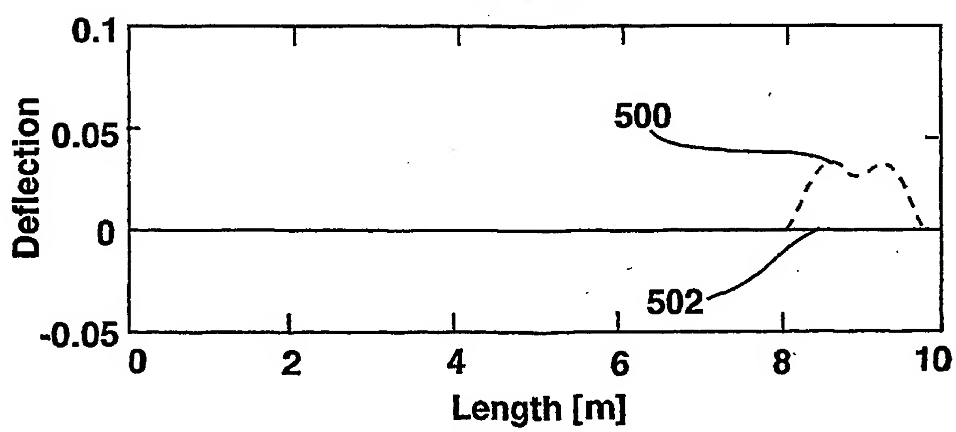
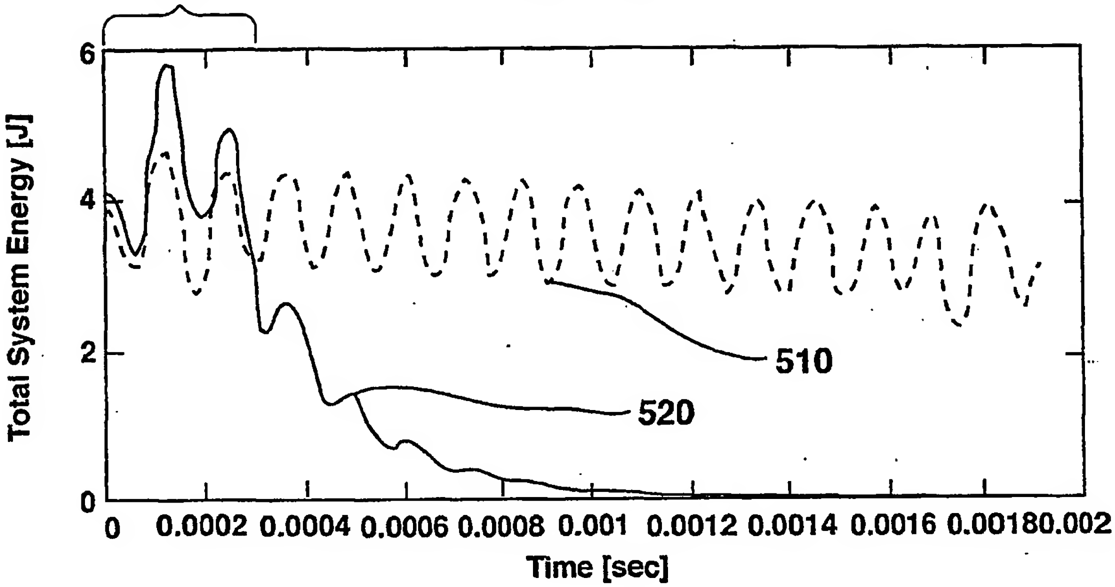


FIG. 10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4
2
0
0 0.0002 0.0004 0.0006 0.0008 0.001 0.0012 0.0014 0.0016 0.00180.002
Time [sec]
510
520